

STUDY OF ASYMMETRY CONFIGURATION
EFFECTS ON PLATE HEAT EXCHANGER PERFORMANCE

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A Thesis Submitted in Partial Fulfillment of the Requirements
for the Degree of Master of Engineering in Mechanical Engineering

Department of Mechanical Engineering

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การศึกษาผลกระทบจากการติดตั้งอุปกรณ์ที่ไม่สมมาตร
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เครื่องแลกเปลี่ยนความร้อนเป็นอุปกรณ์ที่ใช้อย่างแพร่หลาย โดยการศึกษาที่ผ่านมาพบว่า
รูปแบบการไหลส่งผลต่อประสิทธิภาพการถ่ายเทความร้อน ซึ่งก็ได้มีการศึกษาผลจากการ
เปลี่ยนแปลงรูปแบบการไหลด้วยวิธีการต่างๆ แต่ยังไม่มีการศึกษาการเปลี่ยนแปลงรูปแบบการไหล
ด้วยการเปลี่ยนพื้นที่หน้าตัดการไหลให้ไม่สม่ำเสมอตลอดการไหล ซึ่งการศึกษาครั้งนี้มุ่งผลจากการ
เปลี่ยนพื้นที่หน้าตัดการไหลให้ไม่สม่ำเสมอตลอดการไหลด้วยการสร้างติดตั้งอุปกรณ์แบบไม่สมมาตร
ในรูปแบบต่างๆ ซึ่งได้ผลการทดลองแบ่งเป็นสองรูปแบบคือการติดตั้งอุปกรณ์แบบไม่สมมาตรแบบ
ส่งผลต่อสัมประสิทธิ์การถ่ายเทความร้อนจะเพิ่มสัมประสิทธิ์การถ่ายเทความร้อน 6.54% เพิ่มจาก
แบบสมมาตรโดยการทำให้การพาความร้อนดีขึ้นและอีกแบบคือการติดตั้งอุปกรณ์แบบไม่สมมาตร
แบบไม่ส่งผลต่อสัมประสิทธิ์การถ่ายเทความร้อนซึ่งไม่มีผลที่แตกต่างจากแบบสมมาตรปกติ

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Pearachad Chartsiriwattana : STUDY OF ASYMMETRY CONFIGURATION EFFECT ON PLATE HEAT EXCHANGER PERFORMANCE. Advisor: Asst. Prof. Saran Salakij, Ph.D.

Plate heat exchanger is widely used in engineering application. Previous studies indicated that flow profile effects to plate heat exchanger overall heat transfer coefficient with various factors that manipulate flow profile. This study aims to observe the effect of manipulation flow by creating non-uniform cross-section area along with the flow by applied asymmetry configuration to plate heat exchanger. The experimental results categorize into two groups. The first group is an asymmetric configuration that enhances the overall heat transfer coefficient by 6.54% compared to symmetric configuration due to promoting heat convection. The other group is an asymmetric configuration that ineffective to overall heat transfer coefficient compared to symmetric configuration.

Field of Study: Mechanical Engineering

Student's signature.....

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Chapter 1. Introduction

1.1. Introduction to plate heat exchanger

Plate heat exchanger is a heat exchanger device designed based on the concept that allows two fluid to exchange heat thru flat rectangular area which allows fluid to have more heat transfer area. The plate heat exchanger was assembled by stacking plate. At each corner in each plate, there are punched whole which allows fluid to flow all over the corner in each plate in both sides. Normally, rubber gasket was inserted between plate in order to restrict fluid not to leak out and to prevent hot and cold fluid to mixed with each other as shown in Figure 1. In Figure 1, assume hot fluid entrance in the upper right corner. Hot fluid could flow on first and third plate surface but in second and last plate, there was a gasket perimeter block hot fluid flow not to mix with the cold fluid. Lastly, plates and gaskets were tightened with a bolt to the metal frame before it was installed to the operating site.

With this design of plate heat exchangers, all plate heat exchanger has high

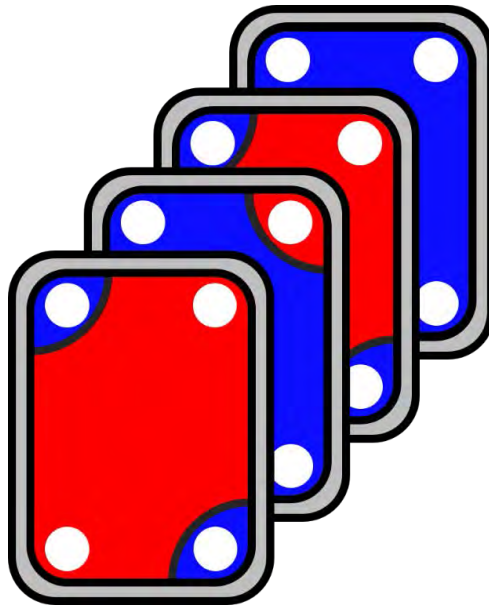


Figure 1 Plate heat exchanger schematic red fluid refers to hot fluid and blue fluid refer to cold fluid

heat transfer area packed in smaller installation area compared to other heat

exchangers at same heat transfer area. Thus, plate heat exchanger can operate with higher thermal efficiency and can draw outlets temperature close to each other. Moreover, with the flexibility of plate heat exchanger, an operator can untighten plate heat exchanger bolt and customize plate heat exchanger by add or remove the plate in device or operator could maintenance by replacing unfunctional plate easily. Later, the operator can tighten the bolt back and the plate heat exchanger will ready for operation. The benefit of the plate heat exchanger is an appropriate choice for engineers to use plate heat exchanger in their process. Plate heat exchangers have been used wildly spread in various industry such as energy generation, chemical plant, and food processing process [1].

1.2. Motivation

According to the literature review in chapter 2 of this thesis, a previous study had investigated on heat transfer mechanism in the plate heat exchanger. Previous studies had shown many interesting factors that effect to plate heat exchanger heat transfer performance, in this thesis will describe major interesting factor that effect on plate heat exchanger heat transfer performance which is flow range and corrugation pattern.

In summary, Tsai et al. [2] and Liu et al. [3] studies have shown friction factor that occurs between plates channel compare to Reynolds number. Both results show that friction factor characteristic can be separated into two groups by a transition range of Reynolds number. The characteristic of friction factor that below transition range linearly decreasing follow increasing of Reynolds number. On the other hand, the characteristic that above transition range cluster and converges as increasing of Reynolds number. These two characteristics depend on Reynolds number which varied form fluid velocity and plate channel.

For plate corrugation geometry which will act as the geometry of channel between plate gap, the corrugation geometry function approximate as flow guide vane

which can change flow direction. Plate geometry has a complex relation to plate heat exchanger heat transfer performance because the effects were compound of various sides of geometry. Each side of geometry has it owns mechanism that effects to plate heat exchanger heat transfer performance.

It can be noticed that both the mechanism of the two factors changes the fluid velocity profile inside the gap channel between plates. The change of fluid velocity profile from both factors have a significant effect on plate heat exchanger heat transfer performance. These mechanisms lead to a concept that, if the fluid velocity profile was changed, heat transfer performance of plate heat exchanger might affect the change.

In this study would like to focus on the effect of changing fluid velocity profile with changing plate heat exchanger geometry that had no previous study focused on. This study proposes to change the fluid velocity profile by modifying a channel cross-sectional area to be non-uniform along streamline. The non-uniform cross-sectional area can be adjusted at plate length each side unequally as asymmetry configuration.

1.3. Objective

This study aims to compare a heat transfer performance of plate heat exchanger with an overall heat transfer coefficient between symmetry configuration and asymmetry configurations from experimental results. This study also compares heat transfer mode inside plate heat exchanger with thermal resistance analysis and extrapolate the result of this study to other plate heat exchanger with dimensionless correlation.

1.4. Scope

The result of this study will focus on experimental result of crossed corrugated commercial plate heat exchanger that feed liquid water at $0.6 \text{ m}^3/\text{hr}$ at both side but at hot side will control inlet temperature at 40°C - 70°C and at cold side will be feed with ambient temperature.

Chapter 2. Literature review

All the previous studies were conducted under the configuration that plate heat exchanger were set up with symmetrical geometry. There is not exact relate previous literature that focus on effect of asymmetry plate heat exchanger on plate heat exchanger heat transfer performance. However, the asymmetry configuration of plate heat exchanger could affect to plate heat exchanger heat transfer performance as same as following factor.

2.1. Effected parameter on plate heat exchanger performance

Many researchers had investigated on the parameter that effect on plate heat exchanger heat transfer performance by all the study's results was categorized and explain as following

2.1.1. Plate heat exchanger configuration

Plate heat exchanger configuration can be setup before plate heat exchanger was operated. This factor will consider designing a plate from the manufacturer. Plate heat exchanger manufacturer could design plate heat exchanger to meet the operating requirement. Some part of the plate such as metal frame and plate rubber need to choose precisely before manufacturing those part. Conversely, plate inside plate heat exchanger can be easily added, removed or replaced by an operator to change plate heat exchanger heat transfer performance. According to Gut & Pinto [4] study had group major configuration that effects to plate heat exchanger performance as detail in following.

2.1.1.1. Fluid channel number

A space between two plates acts as a fluid flow channel. A number of a fluid channel can be added or removed by simply adding or removing plates from the plate heat exchanger. Adjusting plate heat exchanger channel number changes heat transfer area directly and changes fluid flow passage in the plate heat exchanger.

other as a parallel flow but if fluid enters at the opposite side, the flow will counter each other.

2.1.1.4. Passage in channel

This parameter represents the path of fluid inside the channel that was determined by gasket shape. There are two types of passage inside the channel which are diagonal flow and vertical flow. Diagonal flow allows fluid to flow across the plate from one vertical side to another vertical side. On the other hand, diagonal flow let fluid flow from the same vertical side.

Each configuration might not have a direct impact directly to plate heat exchanger heat transfer performance but when those effect of configurations were superposition together can significantly impact heat transfer performance. Besides, Gut & Pinto [5] also optimized the best efficient plate heat exchanger configuration at the same heat transfer area.

On the perspective of this study that focuses on the effect of fluid velocity, its can be seen that almost of configuration have a direct or indirect effect to fluid velocity profile whether it be changing fluid flow passage and change velocity direction. Those changes affect to plate heat exchanger heat transfer performance as a result that Gut & Pinto [5] can change the configuration to optimize plate heat exchanger heat transfer performance.

2.1.2. Flow characteristic

Laminar and turbulent Reynolds ranges are specified by conduit shape, by each shape will have their own unique laminar and turbulent regime. Liu et al. [3] and Tsai et al. [2] had studied from an experimental and numerical result in Figure 3. The results show that at the friction factor behaves in two trends at different Reynolds number. For Reynolds number below 300, friction factor would decrease linearly as Reynolds number increasing. On the other hand, if Reynolds number is higher than 600, overall

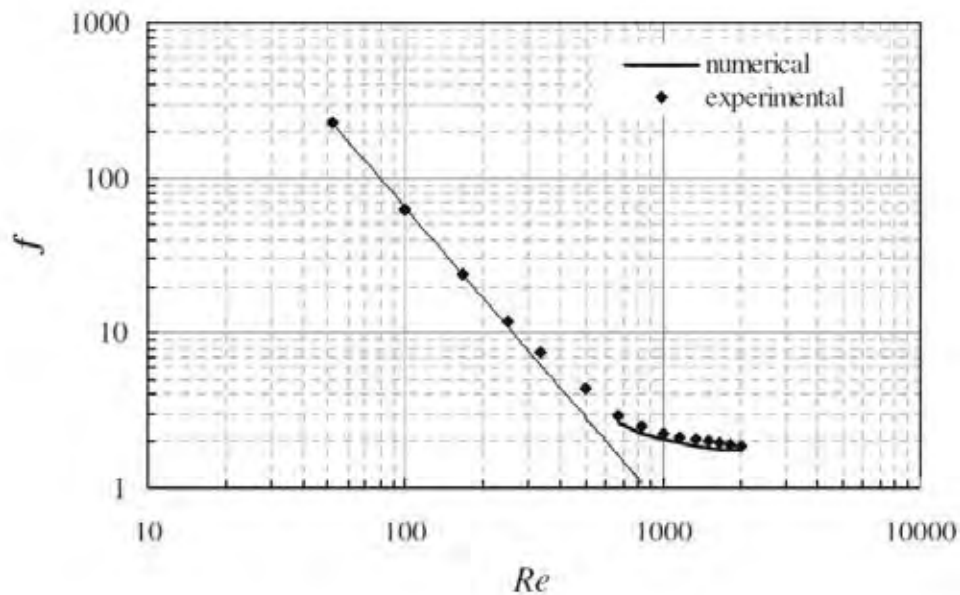


Figure 3 Overall friction factor vs. Re from the numerical model and the experimental result (ref. Liu et al. [3])

friction factor trends to convert down. This can be concluded that inside plate heat exchanger at Reynolds number below 300, fluid behaves as laminar flow and at Reynolds number higher than 600, fluid would behave as turbulent flow. It can be concluded that Reynolds number had an impact on plate heat exchanger performance by Reynolds number is consideration of fluid properties, fluid velocity and conduit shape.

2.1.3. Flow maldistribution

If plate heat exchangers are set as parallel flow, fluid would flow from inlet to the end of the last plate. While fluid flow to the end, the pressure will distribute fluid all over plate to the outlet conduit. The pressure along the conduit is not constant because increasing friction and changing momentum transfer as shown in the study by Bassiouny & Martin [6]. Since the fluid enter inlet passing through inlet conduit to the end, fluid pressure would rise due to decreasing of momentum until the fluid becomes stationary with no momentum at the end of the plate. The fluid pressure also losses

by friction but are not significant when compare to the pressure rises from momentum. Then the pressure will fall immediately. At outlet fluid, the pressure will drop because of friction loss and with pressure converting into momentum to drive fluid out of the device. Aforementioned from Bassiouny & Martin [6] analysis and Bobbili [7] experiment showed that the pressure drop and maldistribution are varied by plate geometry and dominated by a number of a plate stacked. By stacking more plates, the pressure variation and pressure drop between plate heat exchanger would increase. Variation of pressure also effects to fluid velocity, which causes flow rate between each plate to be maldistribution among the device [2].

2.1.4. Corrugation pattern geometry

As the commencement concept of plate heat exchanger that intent to improve heat exchanger heat transfer area, a corrugated pattern was stamped into a plate in order to create ridges surface allows the heat exchanger to have more heat transfer area than a flat surface. Furthermore, these corrugated patterns also impact to plate heat exchanger heat transfer performance and fluid flow velocity profile with the following factor.

2.1.4.1. Inclination angle

The typical design of plate heat exchanger nowadays is a plate with a corrugated pattern because it can maximize heat transfer area and the manufacture also stamp corrugation pattern with an inclination angle. With the inclination angle, the corrugated pattern not only provides more heat transfer area but also function as a guild vane. The inclination angle can act as guild vane because normally the corrugated pattern has furrow like shape, the inclination angle can turn the furrow which will change some of the fluid flow direction along furrow direction.

Focke et al. [8] and Dović & Švaić [9] studied on the cross-corrugated plate heat exchanger which the arrangement of the adjacent plate will turn upside down. Turning the plate upside down will cause the corrugation pattern of the adjacent plate to cross

each other which will be described of this factor in the next section. The results show that the mechanism of inclination angle on plate heat exchanger heat transfer performance by the mechanism of the inclination angle. The inclination angle that was mention in these studies will describe as an angle above the main flow direction which the main flow in both studies is a vertical direction.

The effect on the inclination angle can be explained by the law of momentum conservation at the cross-fluid interface. The cross corrugated pattern creates a fluid interface area which two fluid sub streams from different furrow crossing each together. The result could be observed as the effect of small and large inclination angle with a different result.

2.1.4.1.1. Small inclination angle

For inclination angle that below 45° , most of the momentum remains in its corrugated furrow components and some of the momentum is separated to the other furrow components. The flow at the interface between two streams which are cross direction would be less dominated than the flow at the deeper furrow. The fluid would flow continue their furrow and would reflect at the edge as shown in Figure 4.

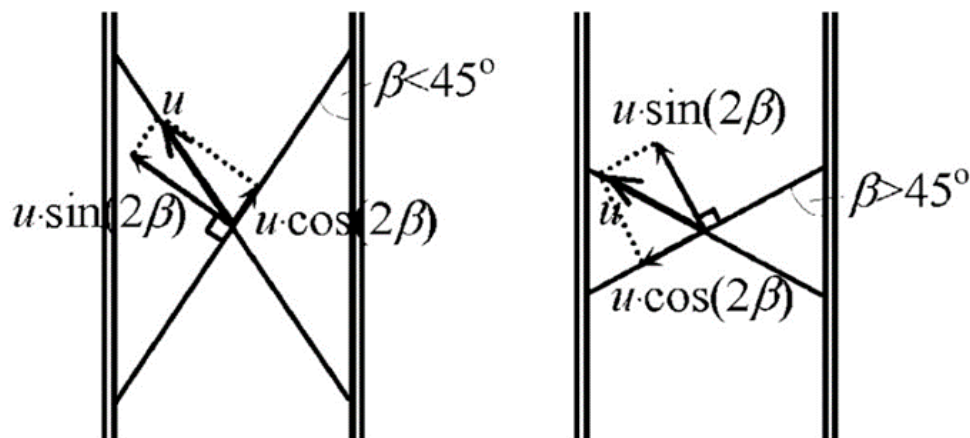


Figure 4 Velocity component in the corrugated furrow, left: small angle and right: large angle (ref. Dovic and Svaic [9])

2.1.4.1.2. Large inclination angle

If the inclination angle is higher than 45° , most of the momentum is separated by the opposite furrow components and some of the momentum remains in its direction. The flow at the interface area dominantly affects the flow in the furrow to separate and flow into the opposite flow like zig-zag pattern as shown in Figure 4.

The study result of Dović & Švaić [10] had developed a correlation between inclination angle and dimensionless heat transfer quantities. The correlation predicts the trend of heat transfer performance which tends to increase when the inclination angle is increased until the inclination angle pass over optimal angle, the heat transfer performance tends to decrease even though the inclination angle is increasing. So, the correlation can specify the optimal inclination angle range at 60° - 70° . The optimal range can be narrow down when considering the effect of aspect ratio which will be detailed later.

In summary, the turning of furrow direction by adjusting the corrugation inclination angle cause flow to change velocity profile. The change of fluid flow velocity profile also affects on plate heat exchanger heat transfer performance as a result of the correlation but at only some optimal inclination angle. The most optimal inclination angle might be the best balance point between the effect of small and large inclination angle at 60° - 70° .

2.1.4.2. Contact point

As mention in the previous section, the cross of the corrugated pattern has its effect on plate heat exchanger heat transfer performance. When two corrugated plates are aligned cross to each other, the edge of corrugate furrow will contact a furrow from another plate cause a contact point at each contact. One furrow edge will contact multiple edges from the other edge form multiple contact point along one edge. In overview, all corrugated furrow edge in one plate will create multiple contact point all over the plate surface.

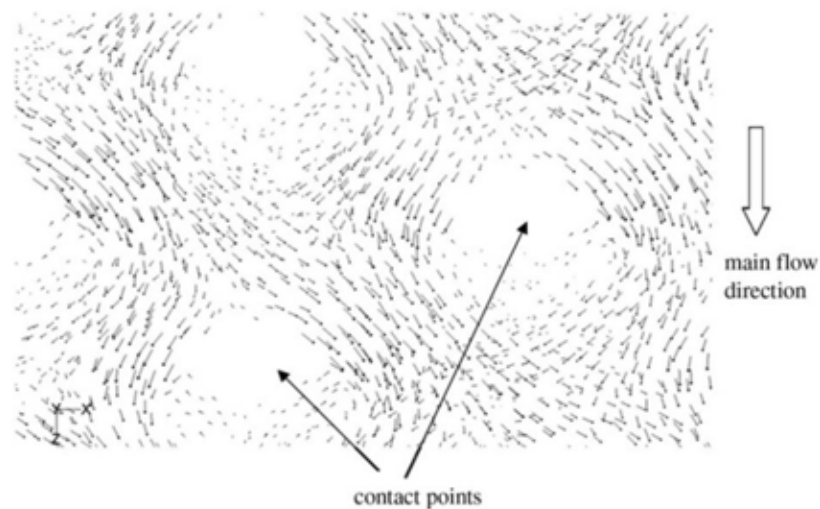


Figure 5 Velocity contour circulation (ref. Liu et al. [3])

Liu et al. [3] result shown the fluid velocity contour profile inside cross-corrugated plate heat exchanger as shown in Figure 5. The velocity profiles in channel seem to circulate around the contact point. The swirling circulation is caused by the crossing of two sub fluid in a furrow, exchange momentum with each other. The fluid will circulate at contact point all over the surface which can cause fluid to change its direction from main flow direction which also effects on plate heat exchanger heat transfer performance.

2.1.4.3. Aspect ratio

As also mentioned previously, another factor that also effects on plate heat exchanger heat transfer performance is aspect ratio. Aspect ratio is a ratio between furrow depth and furrow width. Aspect ratio represents the cross-section of the furrow dimension. According to Dović & Švaić [9] study explained the mechanism of aspect ratio effect to flow in the furrow. The mechanism of aspect ratio affect to flow by flow in this study would group to two sub flow by position which are cross sub flow, sub flow that flow near share surface to crossed corrugated furrow and furrow sub flow, sub flow that flows at bottom of the furrow.

The effect of aspect ratio was categorized to depend on the domination of sub flow. The cross-sub flow is dominated when the furrow is shallow or low aspect ratio. The cross-sub flow can be easily exchanged to the adjacent furrow as same as the effect of large chevron angle but less significant. On the other hand, the furrow flow will stay remain at the bottom of their own furrow which has the same characteristic as small chevron angle with also less significant. Even though the effect of aspect ratio is less than the chevron angle but when superposition of two effects, the heat transfer performance of plate heat exchanger was changed significantly.

2.2. Study methodologies

Previous studies had studied various effect on plate heat exchanger performance using an interesting method to approach each study objective. Each method that was used by the researcher had its own advantage to discover interesting phenomenal in a plate heat exchanger. The method that was frequently used previously were 3 methods as following.

2.2.1. Analytical

In order to understand the mechanism inside the plate heat exchanger, Bassiouny & Martin [6] had observed the parameter that effect to flow distribution and pressure drop in the plate heat exchanger. This paper had analyzed on U-type which the fluid flow from the first plate to the last plate and flow back from the last plate to the first plate. The governing equation such as mass balance equation and momentum equation were applied on plate heat exchanger as a control volume. The equation was simplified by assuming assumptions and derived until the equation express the relation of plate heat exchanger performance with characteristic parameter m . By characteristic parameter m are varied by plate geometry and number of stacked plates. After the equation had been solved, the flow rate will increase in intake direction when m^2 is more than zero. On the other hand, when m^2 is less than zero, the flow rate also increases but increase in flow direction instead.

Even though Analytical technique is convenient to predict various type of plate heat exchanger by solving partial derivative equation and the results are in good agreement. But to study further in the plate heat exchanger, the study method needs to be more precise.

2.2.2. Experimental

For better understand plate heat exchanger mechanism, many researchers had used the experimental method as a tool to gain more understanding. The experiment interesting technique may be characterized into 4 main groups as the following.

2.2.2.1. Measuring parameter at inlet and outlet port

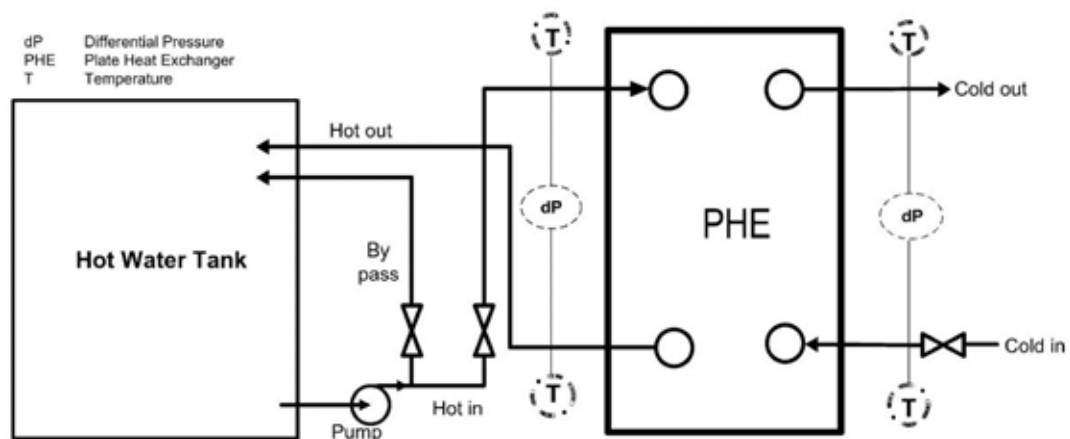


Figure 6 Khan et al. experimental setup (ref. Khan et al. [11])

Khan et al. [11] used experiment technique to study the heat transfer coefficient in multiple plate configuration. In the test rig shown in Figure 6, three plates were used to create two fluid channels, one for hot fluid and other for cold fluid. The hot fluid was circulated by using a variable speed pump to control flow. At first, hot fluid is stored in an insulated tank which temperature is controlled by a resistance temperature detector that connected to a heater to maintain temperature after hot fluid is circulated back from the plate heat exchanger. The cold fluid is not circulated but flow in and drained. The temperature was collected by using RTD at both inlet and outlet from both hot and cold fluid. Pressure drop was also measured by using

pressure transmitters between inlet and outlet from both hot and cold fluid. The flow rate was measured by draining water to empty 5 liter tank at a period time. Temperature and pressure drop from the experiment was used to evaluate overall Nusselt number and overall heat transfer coefficient.

2.2.2.2. Diffusion-limited current technique

Even though the experimental give actual data but are not enough to explain flow characteristic so Focke et al. [8] studied the relationship between plate inclination and overall performance. The study applied DLCT technique to determine heat transfer rate which based on the diffusion-controlled reaction of an ion at the electrode-electrolyte interface. Nusselt number result was calculated from measured viscosity at various inclination angle and Reynolds number.

2.2.2.3. Injected dye visualization

Dović & Švaić [9] investigate on plate geometry on plate heat exchanger performance. In this literature, a flow visualization technique was used by injecting dye into a flow. Test model consists of one plate heat exchanger and thick transparent

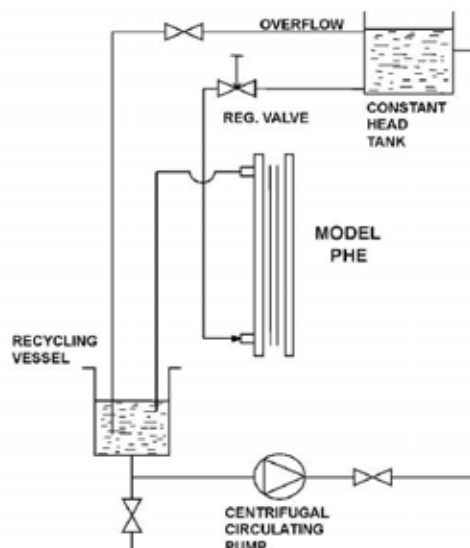


Figure 7 Experimental setup from Damir and Srečko for injected dye technique
(ref. Damir and Srečko [9])

material plate which manufacture with the same geometry as a plate. The test model was fed both hot and cold fluid side at a constant flow rate and constant pressure using a centrifugal pump as shown in Figure 7. The methyl blue was used as a dye in order to visualize the flow. Methyl blue was used because its density is like working fluid and it is clearly visible compared to the working fluid. The flow pattern was recorded by the camera at 1/10 frame per second at various Reynolds number and inclination angle. The collected result is clearly shown in Figure 8.

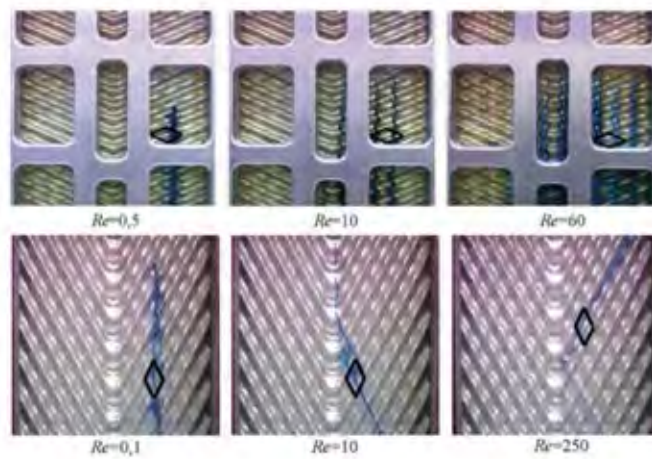


Figure 8 Dyeing result at various inclination angle and Reynolds number (ref. Damir and Srecko [9])

2.2.2.4. Infrared thermography flow visualization

In order to obtain more visualize flow pattern, Kanaris et al. [12] predict flow and heat transfer in a corrugated plate heat exchanger. In this literature, flow local temperature visualization was observed using infrared thermography. The experiment set up was set by assembling two plate heat exchangers where one side is transparent-material plate corrugated pattern to observe local temperature as in Figure 9. The fluid was flow through a plate heat exchanger at a constant temperature for both hot and cold fluid. The temperature field was observed by an infrared thermography camera which captures infrared radiation from solid plate body and 9 thermocouples. It assumes that plate is high conductivity that local temperature at the plate is

approximately the same as local fluid. Figure 10 shows the result from the experiment will present the temperature distribution along at surface of plate heat exchanger.

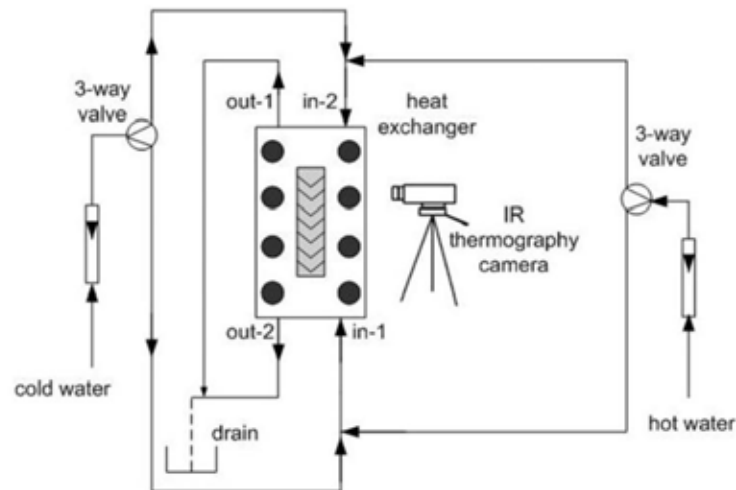


Figure 9 Kanaris et al. Infrared thermography test rig (ref. Kanaris et al. [12])

For all experiment method will provide actual practical information from accurate measurement instrument, the result from the experiment will be reliable and effective for synthesis experiment conclusion. But, for a more expert technique such as injection dye visualization and infrared thermography to obtain more result may cost more expensive.

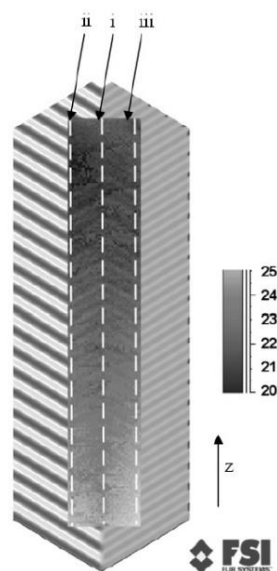


Figure 10 Infrared Thermography result (ref. Kanaris et al. [12])

2.2.3. Numerical model

In order to obtain a more detailed result experiment method with less expense and effort, a numerical model was used numerous times by many studies. The concept of the numerical model is to calculate the governing equation with more precision using a computer. Although the numerical model will result in only the solved equation as a predicted result, the numerical model can provide more detail result with good agreement.

The numerical model methodology starts with giving geometry of interested control volume with the same dimension of an actual control volume. To enhance calculation precision, the geometry will be divided into small volume call mesh so there is multiple mesh compound to form to be the geometry. All the meshes will be assign properties and boundary condition of interested condition. Then all the meshes will be calculated with governing equation one by one. Moreover, a calculated result from one mesh can pass the result to other adjacent mesh so the more mesh divided in geometry, the more precise of the prediction. In addition, all the meshed in all position of geometry can show its result of the calculation which for experimental methodology, some position that cannot install measurement instrument, that position cannot be collect demand value

Bhutta et al. [13] had reviewed the application of numerical model by the review study point that a numerical model is an appropriate tool for predict plate heat exchanger behavior and there is a study that relates to this study as the following.

2.2.3.1. Flat plat heat exchanger numerical model

Galleazzo et al. [14] introduced a numerical method to study simple geometry plate heat exchanger on both parallel and series flow. The experiment set up was alongside to validate the numerical solution. The test rig consists of five stainless steel flat a plate which creates four fluid channels, two for hot fluid and two for cold fluid and runs on various inlets temperature and velocity. Three-dimension model and one-

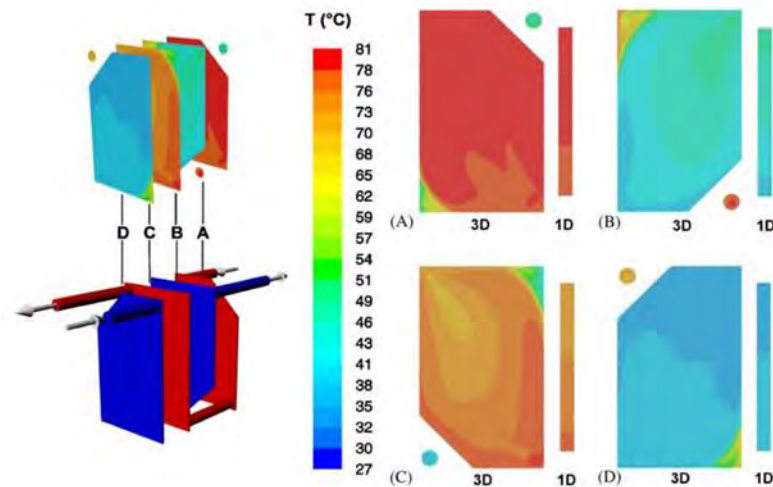


Figure 11 Temperature distribution from the numerical prediction in a flat plate heat exchanger (ref. Galleazzo et al. [14])

dimension model were set up to predict heat transfer. Three-dimension model mesh generation was generated on geometry domain using commercial software GAMBIT. Grid numbers were also reduced until the results were independent with grid number. The model initial conditions such as inlets temperature and velocity were set as same as experiment configuration. Boundary condition also had been set that outlets are pressure outlet. The stainless-steel plate material and water fluid were set to be the same properties as actual properties. The model was solved numerically using the finite element method on FLUENT software. Heat transfer result for both series and parallel flow from the numerical model were overpredicted by 15% because the numerical model did not account for heat loss to an environment which occurs. Three-dimension model also provides all fluid properties in each inner channel as in Figure 11. For one-dimension model predicted heat transfer from reference geometry between Nusselt number, Reynolds number and Prandtl number by those parameters were related with experiment configuration. Heat transfer that was predicted from this model is smaller than experiment result lower than actual experiment results 15% because traditional co-relation was created from previous experimental data pool with statistic methods represent for a general case which might not be precise enough in this case.

2.2.3.2. Corrugated plate heat exchanger numerical model

In order to apply more complex geometry into a numerical model, Kanaris et al. [12] had explored on the potential of using the numerical model to studies plate heat exchanger mechanism by commercial computational dynamics code CFX. In this study, the plate model was constructed with trapezoidal and sinusoidal corrugated shape, but the actual shape is round trapezoidal. The vicinity of boundary condition was filled with prismatic mesh and the other area were filled with tetrahedral mesh. After done grid independence as same as the previous literature, the model was applied with shear stress transport turbulent model. The model was configured with the parameter that is similar to the experiment with some assumptions such as no heat loss to the environment, steady-state flow and constant pressure at the outlet. Figure 12 show Temperature distribution predicted from the model which when compare the result to the previous data, the comparison is almost similar.

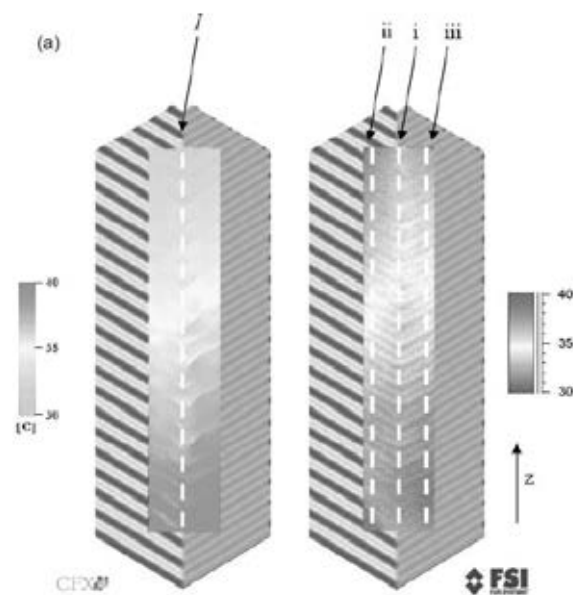


Figure 12 Temperature distribution comparison between the numerical model and IR thermography (ref. Kannaris et al. [12])

2.2.3.3. Unitary cell numerical model

To obtain more detailed fluid flow pattern, Zhang & Che [15] investigated the influence of corrugation profile on the thermal hydraulic performance of cross corrugated plate heat exchanger. In this study, the geometries were created an only single unitary cell which is the partial pattern part of the conduit between two plate heat exchangers. The cells were constructed with a various shape such as sinusoidal, triangular, trapezoidal, rectangular and ellipse shape. The grid was generated with high grid density at the inlet, outlet and wall surface compare to central core region grid density. Then the mesh will be applied with the governing equation thus k-epsilon turbulent model by assuming that fluid is incompressible, and the flow is steady. The model was able to predict the flow pattern. The Colburn j-factor and Darcy friction factor results from the model were compared with Focke [8] and the validation was in good agreement.

All numerical method above obviously efficient tool for studying plat heat exchanger performance. The numerical can predict the result within good agreement with actual experiment results. In addition, the numerical model can discover more detailed which are complicate for experiment method to obtain the result. The numerical model is also convenient for geometric study which changes plate heat exchanger geometry which in a practical experiment may require manufacturing new plate heat exchanger.

Chapter 3. Theory

In order to interpret and compare heat transfer result from the study, the following theories and equations will show the requirement variable need to use to achieve the objective of this study.

3.1. Fundamental equation

3.1.1. Conservation of mass equation

The law of conservation of mass is based on a concept that mass cannot be created or demolished. If applied this law on this study which a control volume is a plate heat exchanger, all the mass which is both hot and cold-water flow rate inlet and outlet must be as same as mass inside control volume change per changed time as shown in the equation.

$$\dot{m}_o - \dot{m}_i = \frac{dM_{CV}}{dt}$$

By \dot{m}_o represent the outlet mass flow rate [Kg/s]

\dot{m}_i represent the inlet mass flow rate [Kg/s]

M_{CV} represent mass inside control volume [Kg]

t represent time [s]

By the system is in steady-state, the mass change per time for both hot and cold side must not change which cause right-hand side term to be zero. Then the equation is deducted to a more simple form.

$$\dot{m}_i = \dot{m}_o$$

This equation can be simply explained that the inlet mass flow rate must be equal to the outlet flow rate for both hot and cold side. Which the inlet and outlet mass flow rate can be calculated from the volumetric flow rate and fluid density.

3.1.2. Conservation of energy equation

The conservation of energy law depends on the basic idea that energy is cannot be created or demolished. So, all energy transfer through a system must be accountable under the conservation of energy law under control volume. By all energy, that transfer rate in and out the system must remain the same as the change of energy in control volume.

$$\dot{Q} + \dot{W} = \frac{dE_{CV}}{dt}$$

By \dot{Q} refer to heat transfer rate [kJ/s]

\dot{W} refer to work transfer rate [kJ/s]

E_{CV} refer to energy stored in control volume [kJ]

After applying the equation to plate heat exchanger system, Heat transfer terms refer to heat loss and internal energy within both hot and cold water and work term only refer to fluid work which pushes fluid in and out the system. According to the system is steady and neglect potential energy from gravitation, the equation will be rewritten as the following.

$$-\dot{Q}_{loss} + \sum \dot{m}_i \left(h + \frac{v^2}{2} \right)_i - \sum \dot{m}_o \left(h + \frac{v^2}{2} \right)_o = 0$$

From the equation, Heat loss and fluid energy outlet flow must balance with fluid energy inlet flow. This equation can calculate heat that is transferred from one fluid to another fluid.

3.2. Overall heat transfer coefficient

Heat transfer rate that was transferred from higher temperature fluid to lower fluid can be evaluated from the conservation of energy equation above. If the temperature difference between inlet and outlet temperature are the same in all cases, the heat transfer rate can be used directly to compare heat transfer efficiency

but the temperatures are not the same so in order to compare heat transfer rate in each case, heat transfer rate must compare in term of heat transfer coefficient which is heat transfer rate at temperature different in log mean temperature difference terms.

$$UA = \frac{\dot{Q}}{LMTD}$$

By UA refer to overall heat transfer coeffect [W/K]

$LMTD$ refer to log mean temperature different [K]

3.3. Thermal resistance analogy

For better understanding in heat transfer mechanism inside the plate heat exchanger, the thermal circuit analogy was used to visualize the heat transfer mechanism inside the device. Comparing heat transfer concept to Ohm's law, heat can refer to an electrical current which flows from higher to lower temperature as same as the electrical current which flows from higher to lower voltage thru heat transfer mode which can refer to electrical resistance. In each heat transfer modes will be redefined as thermal resistance that was compatible in thermal circuit calculation as following.

3.3.1. Heat convection resistance

$$R_{conv} = \frac{1}{hA}$$

By R_{conv} refer to thermal convention resistance [K/W]

h refer to convective heat transfer coefficient [Wm²/K]

A refer to heat transfer area [m²]

3.3.2. Thermal conduction resistance

$$R_{cond} = \frac{L}{kA}$$

By R_{cond} refer to thermal conduction resistance [K/W]

L refer to the length of conducted material [m]

k refer to thermal conductivity [W/mK]

A refer to heat transfer area [m²]

3.4. Dimensionless correlation

To apply the result to other plate heat exchanger that has no same dimension as in this study, all parameter must be calculated as dimensionless quantities and developed relationship between two dimensionless quantities. Then the relation can be applied to other plate heat exchangers.

3.4.1. Reynolds number

The Reynolds number is the ratio between the inertial force to the viscous force of the fluid flow. The Reynolds number represents flow patterns which can be group as laminar, transition and turbulent flow regime. At low Reynolds number flow tend to have smooth streamline which can be called as laminar flow. On the other hand, at higher Reynolds number, the flow tends to have disturbance, swirling pattern which can be called as turbulent flow. The transition flow regime is a regime that was between laminar and turbulent regime which transition's Reynolds number will separate laminar and turbulent flow at various value depend on fluid property, velocity and flow cross-section area.

$$Re = \frac{\rho v d_h}{\mu}$$

By Re refer to Reynolds number

ρ refer to fluid density [kg/m³]

v refer to fluid velocity [m/s]

d_h refer to hydraulic diameter [m]

μ refer to a dynamic viscosity of fluid [Pas]

3.4.2. Nusselt number

Nusselt number is a ratio of convective to conductive heat transfer in fluid flow. Nusselt number represents a proportion of heat convection along with the fluid flow and heat conduction from the fluid contact surface. Normally Nusselt number must higher than one because heat convection must greater than heat conduction.

$$Nu = \frac{hd_h}{k}$$

By Nu refer to Nusselt number

h refer to convective heat transfer coefficient [Wm^2/K]

d_h refer to hydraulic diameter [m]

k refer to thermal conductivity [Wm/K]

As can be seen in this chapter, the theories and equations have shown the requirement variable that needs to be collect and calculate in order to approach this study objective.

Chapter 4. Methodology

As proposed previously, this study objective aims to compare the overall heat transfer coefficient of symmetry and asymmetry plate heat exchanger configurations. In addition, with compare heat transfer mode inside the plate heat exchanger and developed correlation that can be used for other plate heat exchanger. In the previous chapter has identified the required value which needs to collect from a reliable methodology. The methodology must also provide an accurate result for the interpreting process can be done with least imprecisions as possible.

4.1. Methodology selection

The methodology used in this study does not only provide reliable and accurate result but also need to be appropriate in another perspective such as operating time and economical cost. The selected methodology in this study considered all methodology choices based on all perspective which will be explained below. From this study literature review, there was an interesting methodology that can be used which are numerical model and experiment methodologies.

4.1.1. Numerical model

A numerical model is an efficient tool to predict a very detailed result. The numerical model is based on the concept of applying governing equation or correlation into a tiny mesh of element divided in interested control volume can solve those equations using the computational program by the tinier control volume, the more precision of the predicted result. However, in this study, the geometry of the corrugated plate was too complex to use commercial computational force to predict the result.

4.1.2. Experimental

An experimental method is a tool to collect the exact result for this study. Even though experimental might cost in building test rig and measurement instrument,

the test rig can be built to get experiment result with support of campus. So, this study will focus on the experimental method.

4.2. Asymmetry plate heat exchanger

Typically, the plate heat exchanger is assembled by stacking plates to create multiple channels for fluid flow. Then the plate heat exchanger is partitioned by a rubber gasket which not only restricts the fluid to not leak out of device but also controls the fluid direction in the plate heat exchanger to flow through only one channel at a time, which is called series flow, or flow through all the channel between plate at once, which is called parallel flow and also keep both fluids separate from each other. Lastly, the plate heat exchanger will tighten up with bolts to stack plate and gasket together. The symmetric plate heat exchanger bolt must be tightened until the lengths of tightening bolts are equal to each other. When all sides are equal, all



Figure 13 KAUDY, M2 plate heat exchanger installed in a test rig

channel areas uniform along the channel, in which fluid can flow with the same velocity along the channel.

In this test rig commercial plate heat exchanger from KAUDY, model M2 is used for observing the difference. Model M2 heat exchanger is a compact size with 0.0373 m^2 heat transfer area and dimension of 47 cm length, 10 cm width and 0.5 mm thick with 1 cm furrow width and 3 mm in depth. Plates are stamped with horizontal chevron corrugated pattern with 60° refer to the main flow. The inlet and outlet ports are placed at the same side, the hot fluid inlet is placed over the outlet and the cold fluid inlet is placed under outlet as shown as an arrow in Figure 13 and Figure 14.

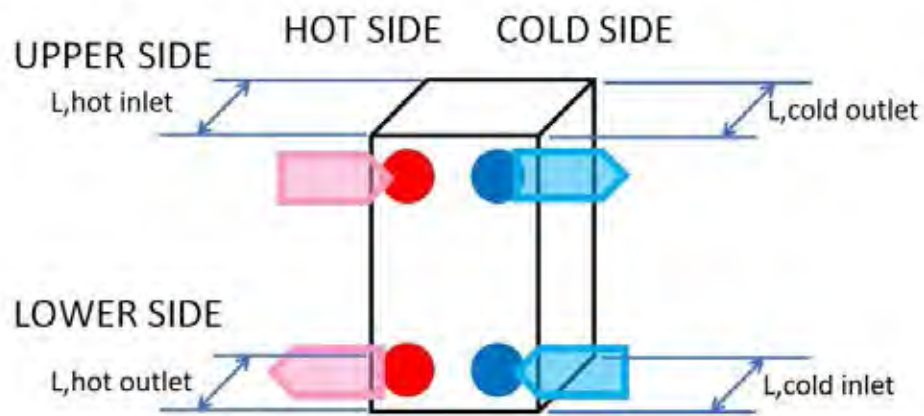


Figure 14 KAUDY, M2 plate heat exchanger symmetric configuration schematic

In this study, asymmetry plate heat exchanger will be defined as the plate heat exchanger, in which the flow cross-section area is varied along with the flow. Theoretically, the flow behavior can be adjusted by altering the flow cross-section area, by decreasing the area to accelerate flow or increasing the area to decelerate the flow. In this study, we focus on investigating the effect of altered channel cross-section area along with the flow, as a result of the asymmetric shape of the plate heat exchanger, to the overall heat transfer performance.

In order to create non-uniform cross-section areas along the channel, the length at each side will be adjusted. Plate heat exchanger length at upper (upper hot and cold) and lower (upper hot and cold) side in Figure 15 must differ from each other. If the upper length is shorter than lower length, the hot fluid channel will shape like a diffuser and the cold channel will shape like a nozzle. For the other case, if the

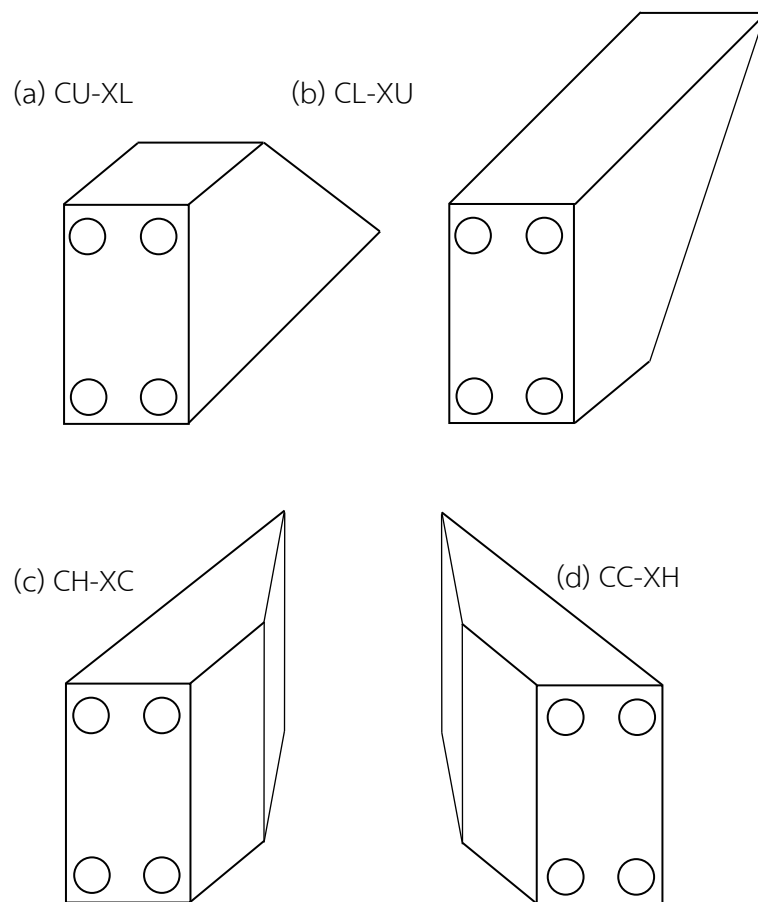


Figure 15 Asymmetry plate heat exchanger configurations

lower length is shorter than the upper length, the hot fluid channel will shape like a nozzle and the cold channel will shape like a diffuser.

The result of the experiment consists of five configurations. First two are configurations that alter the upper and lower length, one is compressing the upper (both hot and cold) length and expanding the lower (both hot and cold) length (CC-

XL) as shown in Figure 15 (a), and another configuration compressed the lower length and expanded the upper length (CL-XU) as shown in Figure 15 (b).

And the other two cases are the case that intends to distinguish the effect of non-uniform channel areas between a plate with two configurations. One is to compress the hot side length (both upper and lower) and expanded the cold side length (both upper and lower) (CH-XC) as shown in Figure 15 (c). The other configuration is to compress the cold side length and expanded the hot side length (CC-XH) as shown in Figure 15 (d).

4.3. Test plan

In this experiment will operate with KAUDY, model M2 commercial plate heat exchangers. Feeding in with hot liquid water at 40°C-70°C with ambient liquid water at another side at both 0.6 m³/hr because flow inside channel will remain turbulence inside the channel [3] at all test with controllable heat load as in every case. In practice, the plate heat exchanger exceeds its maximum asymmetry length without leaking fluid by all geometry as shown in *Table 1*. Then the dimension of asymmetry length will be switch to the opposite side in order to create opposite asymmetry configuration. Lastly, all length of all asymmetry side from each case will be average to calculate symmetry length. This symmetry length will be applied at all side of the plate heat exchanger as symmetry case that will be used to compare heat transfer coefficient with asymmetry case. All the length that was measured in *Table 1*, was measured at bolt length inside the metal frame that.

Table 1 Plate heat exchanger configurations length

Configurations	Lengths mm. \pm 0.55 mm (Asymmetry percentage)			
	Upper hot	Upper cold	Lower hot	Lower cold
CU-XL	41.17 (4.2%)	41.15 (4.2%)	44.79 (-4.2%)	44.71 (-4.1%)
CL-XU	45.30 (-5.5%)	45.27 (-5.4%)	41.27 (3.9%)	41.22 (4.1%)
CH-XC	39.53 (8.0%)	46.49 (-8.2%)	39.54 (8.0%)	46.48 (-8.2%)

CC-XH	46.36 (-7.9%)	39.50 (8.1%)	46.37 (-7.9%)	39.51 (8.0%)
Symmetry	42.96	42.96	42.96	42.96

4.4. Experimental setup

In order to obtain overall heat transfer coefficient and Nusselt number at each configuration, the plate heat exchanger must be fed with liquid water for both sides at the same condition except plate heat exchanger length. In this study, the flow rate is set to be the same at 0.6 m³/hr, but the hot fluid inlet temperature must be varied from 40°C-70°C in each case. The test rig was set up with the following component specification.

4.4.1. Plate heat exchanger specification

A plate heat exchanger that was used in this experiment has specification as *Table 2*.

Table 2 Plate heat exchanger specification

Plate Heat Exchanger model	KUADY, M2
Plate material	Alloy 306
Heat transfer area	0.03733 m ²
Plate thickness	0.50 mm
Plate width	10 cm
Plate length	47 cm
Corrugated pattern	Horizontal, 60°
Furrow width	1 cm
Furrow depth	3 mm

4.4.2. Hot side

The hot fluid system begins at hot side water storage. The storage will contain hot side fluid and heat hot side fluid up with six 3 kW coil heater until the temperature of hot inlet fluid reaches the required temperature by the control system which will be detailed on other section. Then the hot fluid with require temperature will be feed to plate heat exchanger by a pump. Before the hot fluid enter plate heat exchanger,

some of the fluid will be by-passed and return to hot side water storage because to control the flow rate to remain constant along with the test. After hot fluid enter plate heat exchanger, the hot fluid will be cooled down by plate heat exchanger and return to the hot side water storage for reheat and reuse the fluid again.

4.4.3. Cold side

For the cold side, fluid will be rested at a cold side water storage to adjust its temperature to conditioned room temperature. Then the cold fluid will be feed with pump into plate heat exchanger with a controlled flow rate. The cold side also controls flow rate by bypass some fluid back to the cold side water storage before enter plate heat exchanger. After the cold fluid enters plate heat exchanger and heats up, the cold side fluid will be discharged and not recycle the fluid.

4.4.4. Fluid flow

As the hot and cold side was set up in detailed above. The plate heat exchanger will be feed with hot and cold side fluid as detailed in *Table 3*. Plate heat exchanger fluid configuration also detailed in *Table 3*.

Table 3 Fluid configuration

	Hot side	Cold side
Fluid	Water	Water
Flow rate	0.6 m ³ /hr	0.6 m ³ /hr
Temperature	40°C-70°C	23.7 ± 0.6°C (Ambient temperature)
Channel	5	5
Passes	1	1

4.4.5. Control unit

This experiment must control hot inlet fluid temperature precisely for the most accurate results as possible. The hot fluid side will be heated with six 3 kW coil heater

and the heater will be controlled with two methods which are fine-tuned and baseload to heat the hot side fluid until it reaches the required temperature.

4.4.5.1. Fine-tune

The fine-tune method allows the coil heater to generate heat precisely for accurate the hot inlet temperature. The fine-tune method operation begins at a resistance thermometer (RTD PT100) that install at the heat tank which returns the temperature signal to a thermostat. When the thermostat receives that the temperature at the hot side is lower than the required temperature, the thermostat will return linear output voltage to solid-state relay. When the solid-state relay receives the linear output, the solid-state relay can supply various voltage to 3 kW coil heat heater which can generate heat load between 0-3 kW in order to draw the hot side temperature to the required temperature.

4.4.5.2. Baseload

When the plate heat exchanger is operated at a high temperature which causes more heat load, the fine-tune heater cannot generate enough heat load for the system. The baseload method which connects to five 3 kW coil heater will help fine-tune generate heater until it reaches the required temperature. The baseload method will use on-off to connect two 3 kW coil heater to single-phase input and connect three 3kW coil heater to three-phase input in sequence.

4.4.6. Measurement instrument

According to the requirement data from this experiment, this experiment must provide temperature from hot and cold side at both inlet before enter plate heat exchanger and outlet after fluid have exchange heat with each other This experiment must also provide flow rate of both hot and cold fluid in order to calculate the amount of fluid flow thru plate heat exchanger. The measurement instrument used in this experiment is a tool to collect those require data. The measurement instrument must be installed properly to collect the raw data from the experiment with the most

accuracy and precision. Moreover, those measuring instrument must be calibrated with other reliable measurements to confirm that the measurement instrument can provide the best data for the study. So, in this study, the required data is the temperature from the hot and cold side at both inlet and outlet and flow rate from both hot and cold fluid. The measurement instrument in this study will be installed with detailed in *Table 4*.

Table 4 Measurement instrument specification

Instrument	Flow rate	Temperature
Type	Rotary flow meter	Thermometer, type K read with digital data logger
Resolution	Hot: 10 gal/hr (0.045 m ³ /hr) Cold: 10 l/hr (0.01m ³ /hr)	0.1°C
Location	After outlet port	Hot and cold at before entering inlet and after exit outlet port
Calibration	Filled volume at measured time	Mercury thermometer
Calibrated accuracy	Hot: 98.426% Cold: 97.985%	T _{h,i} : 99.18%, T _{h,o} : 99.81%, T _{c,i} : 99.80%, T _{c,o} : 99.68%,

4.5. Experiment procedure

The experiment procedure of this study starts with selected asymmetry case to study. The selected asymmetry case length will be applied to all side of geometry between bolt as *Table 1*. Then preheat the hot side water from ambient temperature to 40°C, the start temperature. When the hot side temperature reaches the requirement, both of pump will initiate and feed start to feed fluid to plate heat exchanger. The valve of both hot and cold side will be adjusted in order to control flow enter plate heat exchanger to be 0.6 m³/hr for both hot and cold side. After hot and cold fluid pass thru plate heat exchanger the fluid will start to exchange heat with

each other so the hot side inlet which circulates hot water, inlet temperature will drop from 40°C about 7°C-10°C. After all component in the test rig is operated, the thermostat will be set at 40°C and wait until the system stays on steady-state for 1-2 minute. In addition, in order to prevent error, the collected data will collect not only the required temperature but also $\pm 0.5^\circ\text{C}$ around to require temperature. After collecting the required temperature, the thermostat will be set with more 5°C and repeat the collecting data procedure again until the hot inlet temperature reaches 70°C. When all data were collected in one test, the hot side tank must be rested until the hot side inlet temperature return to room temperature and the experiment can start again if the geometry remains the same but if have to change plate heat exchanger geometry, all pump and heater must be stopped to the operator to adjust the plate heat exchanger length.

4.6. Data analysis

After all raw data was collected from the measurement instrument in the experiment set up, the raw data cannot indicate or explain the effect of asymmetry effect on plate heat exchanger heat transfer performance compared to symmetry configuration. The raw data must be analyzed and calculated in another variable with Microsoft EXCEL and MATLAB for more interpret and understand.

4.6.1. Overall heat transfer coefficient

The overall heat transfer coefficient can be calculated as explained in section 3.2 but in detail, the heat transfer rate was calculated from the equation below.

$$\dot{Q} = \dot{m}c_p\Delta T$$

By \dot{Q} refer to heat transfer rate [W]

\dot{m} refer to mass flow rate [kg/s]

c_p refer to specific heat capacity at constant pressure [J/K kg]

ΔT refer to a temperature different [K]

The heat transfer rate in the equation above will be calculated with cold fluid properties because will account only the heat that transfers from hot side to cold side does not account with heat from the hot side to ambient. by mass flow rate will be calculated with cold side flow rate with a density of average cold side temperature as same as the specific heat capacity which is the properties at the average temperature of the cold side. The temperature different used in this equation also a temperature difference of cold side fluid at each point of data.

After the heat transfer rate was calculated, heat transfer will be used to calculate the overall heat transfer coefficient to compare the heat transfer rate with other configuration with the same temperature difference.

4.6.2. Thermal resistance analysis

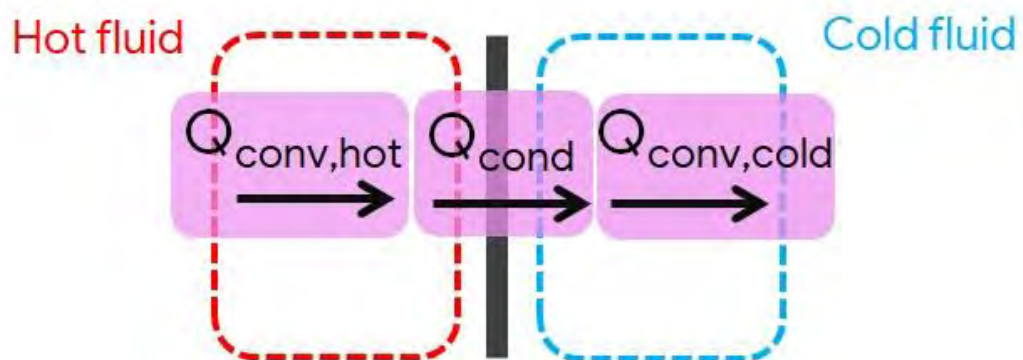


Figure 16 Simplify plate heat exchanger system

In order to analysis the heat transfer mode inside plate heat exchanger with thermal circuit analysis method, the test plate heat exchanger system must simplify to one large plate heat exchanger with a thickness of only one plate and the area of all heat transfer area in the actual plate heat exchanger as shown in Figure 16. Then the thermal circuit can be applied on the simplify system following heat transfer path from convection from the hot side to conduction in plate and end at convection to the cold side. The thermal circuit can be analogized as same as the simplified circuit which the heat transfer from heat convection resistance of hot side to thermal

conduction resistance of plate and end at heat convection resistance of cold side as shown in Figure 17.

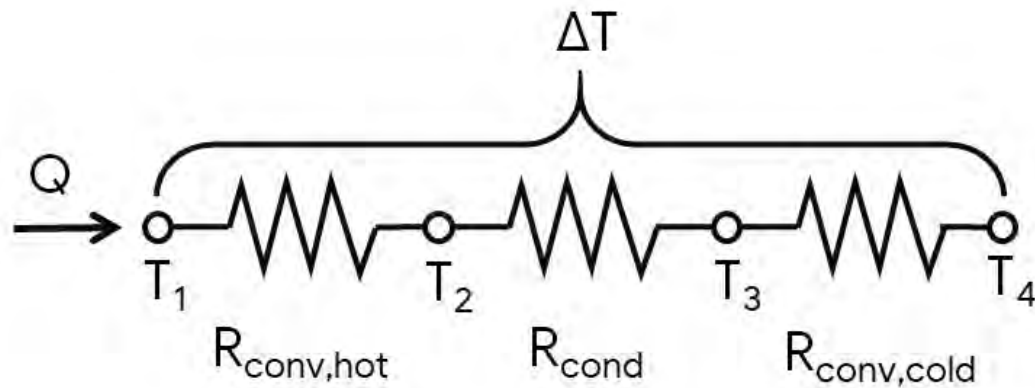


Figure 17 Thermal circuit analogy

After converting the simplified system to the thermal circuit and applied the calculation, not all the collected value cannot solve the equation because cannot determine heat convection coefficient of both hot and cold side. In order to solve two equations, it must assume that Nusselt characteristic of both hot and cold side is the same. So, the heat convection coefficient can be calculated from Nusselt number which is same for both sides, a thermal hydraulic diameter of each side which also the same and thermal conductivity of fluid which is the fluid property of each side. Lastly, when all thermal resistance can be determined, the thermal resistance can be compared and study the heat transfer mechanism inside the plate heat exchanger.

4.6.3. Dimensionless correlation

Dimensionless correlation of plate heat exchanger heat transfer performance can extrapolate the results of this study to other plate heat exchanger. The variables that were proposed to compare the heat transfer performance in correlation are Reynolds number and Nusselt number but with an assumption set previously that both hot and cold side has same Nusselt number characteristic. So, the generalized

Lévêque equation [16] is used to the developed correlation that can explain the effects clearly.

4.7. Error propagation

Even though all collected values from an experiment were actual values, but all the values have their own error from observer and instrument. The error occurs in this experiment was to account for every calculated variable by root square sum of precision error and bias error from the experiment. By The precision error was occur from the repeating of the test, the more repeated test can lower the precision error from student t-distribution and the bias error occur from the accuracy and resolution of measurement instrument which all error is evitable. All error propagation in this study were calculated with MATLAB code shown in Appendix 7.2.

Chapter 5. Result & Discussion

After the experiment was executed and all data were analyzed as a result of this study, the result will be interpreted the physical meaning to explain the effect of the asymmetry configuration on the plate heat exchanger.

5.1. Overall heat transfer coefficient

The overall heat transfer coefficient can be visualized by plotting relation between log mean temperature difference and heat transfer rate by the overall heat transfer coefficient can be noticed by the slope of the relation as shown in Figure 18.

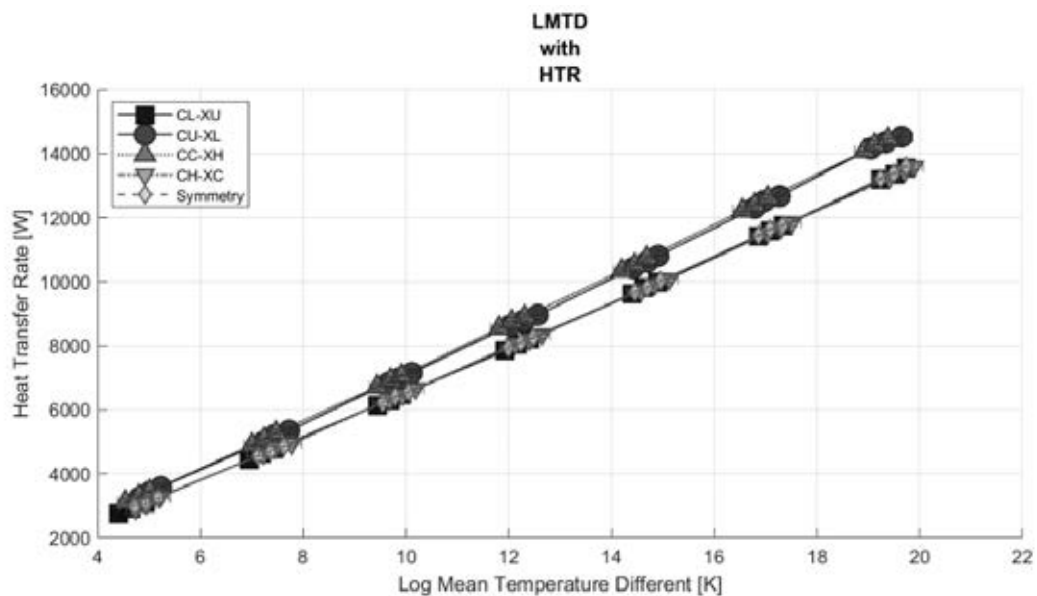


Figure 18 Overall heat transfer coefficients results

The difference in configuration is significantly higher than the error bar. From Figure 18, it can be noticed that the result of each case separates over the result error into two groups compare with the symmetry overall heat transfer coefficient.

The first group, which are CU-XL and CC-XH configurations, have higher slope than the symmetry configuration means that CU-XL and CC-XH configurations have a higher heat transfer coefficient than symmetry configuration. The higher of overall heat transfer coefficient of asymmetry configuration can imply that the asymmetry CU-XL and CC-XH configurations effect to heat transfer coefficient.

The second group, which are CL-XU and CH-XC configurations, have the same slope with the symmetry configuration mean that CL-XU and CH-XC configuration have no different of overall heat transfer coefficient from the symmetry case. The same overall heat transfer coefficient of asymmetry configuration can imply that the asymmetry CL-XU and CH-XC configurations do not affect the overall heat transfer coefficient.

The results of overall heat transfer from asymmetry configuration were affected differently by CU-XL and CC-XH asymmetry configurations can enhance overall heat transfer coefficient but CL-XU and CH-XC configurations have no effect on overall heat transfer coefficient and the overall heat transfer coefficient of each configuration will be detailed in *Table 5 Overall heat transfer coefficient of each configuration*.

Table 5 Overall heat transfer coefficient of each configuration

Configuration	Overall heat transfer coefficient
CL-XU	703.2728 [W/K]
CU-XL	762.8662 [W/K]
CC-XH	765.0335 [W/K]
CH-XC	702.8257 [W/K]
Symmetry	707.7870 [W/K]

5.2. Thermal resistance analysis

For further understanding of the heat transfer mechanism, the thermal resistance from the thermal circuit analogy can identify the heat transfer mechanism inside the plate heat exchanger. The result of the average thermal resistance of each heat transfer mode in each configuration as shown in Figure 19. It shows that thermal conduction in each case is all tremendous lower than both hot and cold heat convection thermal resistance due to the conductivity of the plate have no significant change in all cases.

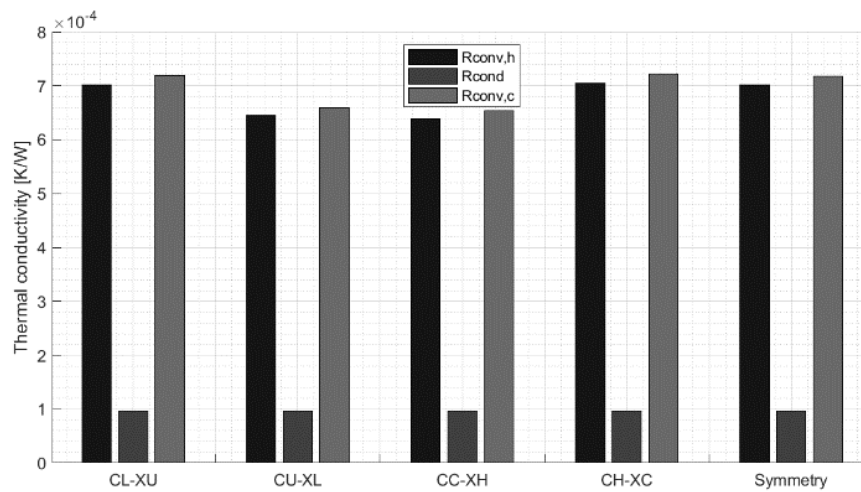


Figure 19 Average thermal resistance of each heat transfer mode in each configuration

If consider only heat convection resistance in the plate heat exchanger, Figure 20 shows that heat convection resistance decreases when the temperature different increase and heat convection resistance for hot side also lower than the cold side which sensible for the increasing of temperature different allow heat to transfer easily cause the thermal resistance to decrease. The result of asymmetry configuration, for both hot and cold side, in the figure also separate itself significantly into two groups by one, CU-XL and CC-XH configurations have lower heat convection resistance than symmetry configuration mean that CU-XL and CC-XH configurations allow heat to convect easier than symmetry configuration and two, CL-XU and CH-XC configurations have same heat convection resistance with symmetry case mean that CL-XU and CH-XC configurations have allowed heat to convect with the same resistivity as the symmetry configuration.

The thermal resistance analysis results also explain the overall heat transfer coefficient result by CU-XL and CC-XH configurations have higher overall heat transfer coefficient and have lower heat convection resistance. It can be implied that asymmetry CU-XL and CC-XH configurations effect on plate heat exchanger by enhancing the overall heat transfer coefficient from enhancing heat convection.

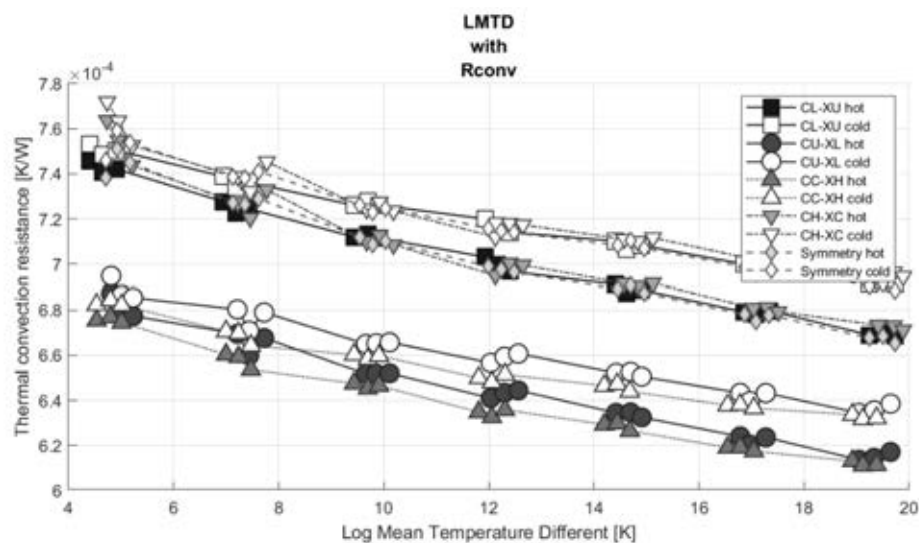


Figure 20 Thermal convection resistance in each configuration

On the other hand, CL-XU and CH-XC configurations have no significant change in overall heat transfer coefficient and no significant change in heat convection. It can be implied that asymmetry CL-XU and CH-XC configurations do not affect the plate heat exchanger for both overall heat transfer coefficient and heat convection.

5.3. Dimensionless correlation

The dimensionless correlation must be developed in order to apply the experiment result to predict heat transfer performance other plate heat exchanger with different geometry. As mention previously in the calculation of thermal resistance, the Nusselt number characteristic was assumed to be same for both hot and cold side so in this correlation with use generalized L  v  que equation [16] to define heat transfer performance with less effect of flow characteristic between hot and cold side by divide by Prandtl number. The correlation was developed base on linearizing the logarithm

experimental result shown in Figure 21 and the constant of the correlation is shown in Table 6, furthermore, accuracy other function can be used to develop the correlation.

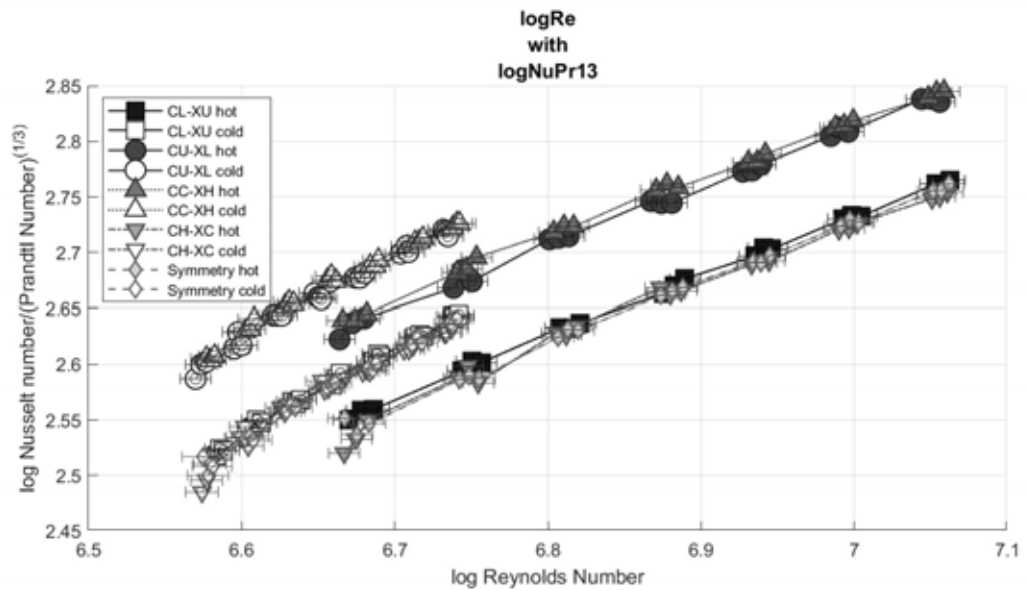


Figure 21 Logarithm experimental dimensionless result

Table 6 Constant for Nusselt number correlation for each plate heat exchanger

$$\text{configuration: } Nu = cRe^m Pr^{1/3}$$

Configurations	m	c
CL-XU	0.466	0.589
CU-XL	0.462	0.654
CC-XH	0.459	0.674
CH-XC	0.472	0.560
Symmetry	0.474	0.555

According to the dimensionless correlation result, all configuration results have the same slope but a shift in the vertical axis which reflected by correlation constant. The power of Reynold number (m) after taken log, represents the slope of the results which all configurations have the same m constant. The coefficient of Reynold number (c) after taken log, represent the intercept point with a vertical axis which was not the same for all configuration.

The result of asymmetry configuration interception point with vertical axis can be significantly categorized into two groups by the symmetry configuration results by over the results error. The first group, CU-XL and CC-XH configurations, have higher vertical intercept point than symmetry configuration means that heat transfer performance is higher than symmetry configuration consistent with the previous result. The other group, CL-XU and CH-XC configurations, have no different vertical intercept point between the symmetry configuration mean that heat transfer performance has no significantly different from the symmetry configuration which also consistent with the previous result.

In summary, asymmetry configuration effect to plate heat exchanger heat transfer performance differently by the effect in this study can be categorized into two groups which are one, the asymmetry configuration (CU-XL and CC-XH) that can enhance plate heat exchanger overall heat transfer efficiency 6.54% by increase heat convection in plate heat exchanger and two, the asymmetry configuration (CL-XU and CH-XC) that not effect to plate heat exchanger heat transfer coefficient.

5.4. Discussion

The interesting issue from the result is that only asymmetry CU-XL and CC-XH configurations can enhance plate heat exchanger heat transfer coefficient but in contrast, asymmetry CL-XU and CH-XC configurations should decrease plate heat exchanger heat transfer coefficient but actually the result shows that the CL-XU and CH-XC configurations do not affect heat transfer performance.

The result explains that the enhancement of the heat transfer coefficient is a result of an enhancement of heat convection. So, for the case the asymmetry configuration enhances the heat transfer coefficient, the asymmetry length manipulates flow profile to enhance heat convection. On the other hand, the asymmetry configuration that does not affect to heat transfer coefficient, the

asymmetry length might not manipulate or manipulate flow profile, but the manipulation did not affect to heat convection absolutely.

The most reasonable explanation for the issue is different of asymmetry length. According to this study result, the common asymmetry length among the asymmetry configuration that enhances heat transfer coefficient is a compression of cold outlet side and expansion of hot outlet side which only change of this side can enhance heat convection.

The other common asymmetry length among the asymmetry configuration that does not affect to plate heat exchange heat transfer coefficient is compression of hot outlet side and expansion of cold inlet side. The result from the asymmetry configuration that does not affect the heat transfer coefficient is not altered from symmetry configuration. So, it cannot imply any discussion further because the scope of this study cannot identify whether the change of asymmetry length has no effect to flow profile at all or the change of asymmetry length also effect to flow profile but heat transfer was refute each other so the heat transfer coefficient has no different from the symmetry configuration.

For further explanation, the effect that only some asymmetry can enhance the heat transfer coefficient, but another asymmetry configuration does not affect. In addition, Han et al. [17] study that can display temperature distribution inside plate heat exchanger which installed with symmetry configuration. From Figure 22, the temperature distribution in plate heat exchanger channel shows that the area that heat can highly transfer is the area that has a high temperature difference between two plates and the area that heat can hardly transfer is the area that has a low temperature difference between two plates.

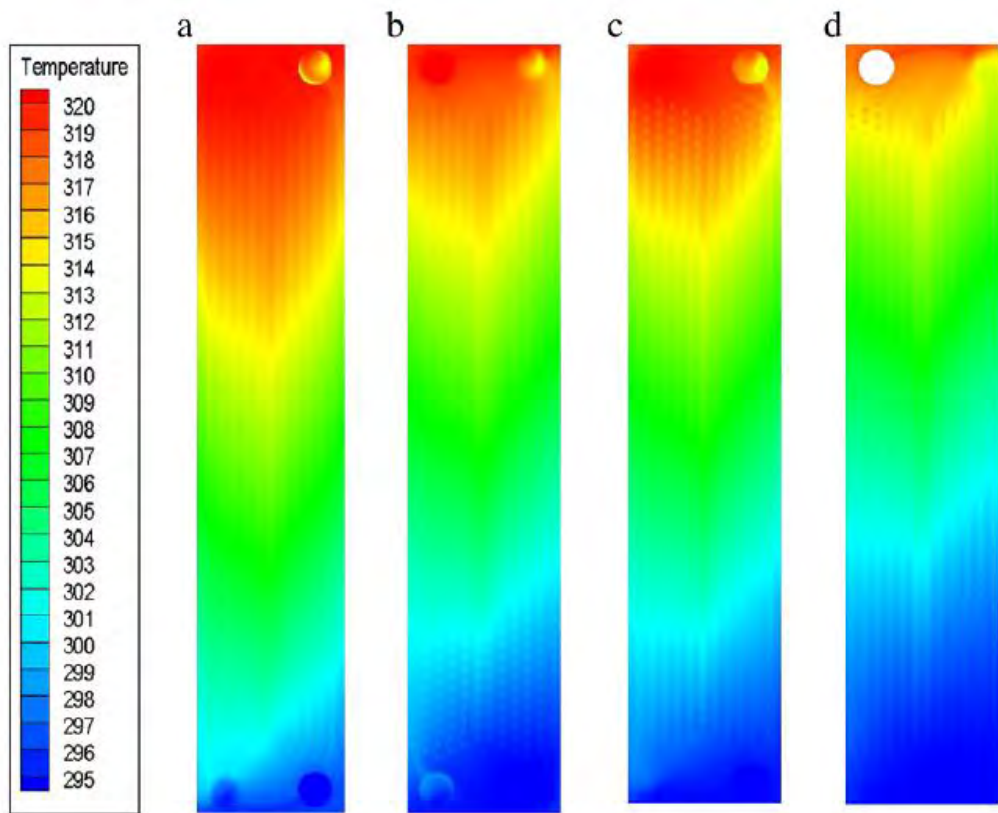


Figure 22 Temperature distribution from the numerical model (ref. Han et al [17])

This visualization can help explain the heat convection effect in this study that the change of geometry that effect on plate heat exchanger heat transfer coefficient might change the temperature difference between two plates and allow more heat to convect in the plate heat exchanger. This explanation needs more evidence to support the mechanism of heat convection such as actual temperature distribution for asymmetry plate heat exchanger configurations.

Chapter 6. Conclusion

6.1. Asymmetry configuration

Asymmetry configuration effect on plate heat exchanger heat transfer coefficient differently in each configuration setting. Asymmetry CU-XL and CC-XH configuration can enhance the overall heat transfer coefficient to 6.54% by promoting more heat convection. But, asymmetry CL-XU and CH-XC configuration that ineffective to overall heat transfer performance and does not promote heat transfer.

6.2. Dimensionless correlation

For the constant in dimensionless was shown in *Table 7*.

Table 7 Average constant for Nusselt number correlation for each plate heat exchanger configuration: $Nu = cRe^mPr^{1/3}$

Configuration	m	c
Asymmetry: Enhance heat transfer	0.460	0.663
Asymmetry: Ineffective	0.469	0.574
Symmetry	0.474	0.555

6.3. Further study

In order to discover the effect of asymmetry configuration on plate heat exchanger heat transfer performance, more study must be conducted to explain more mechanism of the effect. The further study can conduct by visualized properties inside plate heat exchanger and study on various asymmetry geometry.

Visualization of properties inside plate heat exchanger can provide more explanation about the asymmetry effect on plate heat exchanger performance, such as in this study that requires more temperature visualization to explain heat transfer mechanism inside the plate heat exchanger.

The visualization of properties inside plate heat exchanger can be conducted by install local sensor between plate such as to inspect temperature distribution inside the plate heat exchanger, the thermometer must be installed between the plate or using an infrared camera to detect the thermal distribution. Another technique used to visualize properties is using a numerical model to predict the properties between plate.

Another study that can conduct is to focus on different shape of asymmetry geometry. As mention in the scope of this study, this study attempts to see an overview of each configuration. The investigation can be conducted on various of asymmetry such as each side of asymmetry length can be varied to compress or expand, moreover, the study can also combine the effect of various side and length.

Chapter 7. Appendix

7.1. Experimental set up

7.1.1. test rig



7.1.2. Hot water storage



7.1.3. Heater



7.1.4. Controller



7.1.5. Pump



7.1.6. Cold water storage



7.2. MATLAB calculation code

```
function [RAW, CAL, ERR] = DataProcess(caseNO)
%% initial set up
disp('Initiate Data Process Protocol')

if ~exist('caseNO')
    disp('caseNO var not exist');
    caseNO = [1:8];
    %caseNO = 1;
end

filename = "NewLocalExperimentData";

if ~( exist('caseNUM') & exist('caseNAME') )
    load('NewcaseNUMcaseNAME');
    %[caseNUM,caseNAME] = DataHead(filename,2);
    disp('complete : DataHead()')
end

n = size(caseNO);
n = n(2);
%% DataFind()
for icase = 1:n
    i = icase;
    istr = num2str( i );
    [ isheet{i},sheetname(i) ] = DataFinder(caseNO(i),caseNUM,caseNAME);
    disp(['DataFind : ', istr])
end
allcasesize = size(isheet);
allcase = allcasesize(2);

%% data collect
for icase = 1:allcase
    disp('Collecting Data')
```

```

i = icase;
subcasesize = size( isheet{i});
subcase = subcasesize(1);
for jcase = 1:subcase
    j = jcase;
    %load data
    sheetNO = isheet{i}(j);
    [dataNUM,dataSTR,dataRAW] = xlsread(filename,sheetNO);

    %[Case] : collect case name
    caseNOstr = strcat( sheetname{i},' ', num2str(j) );
    RAW{i,j}.Case = caseNOstr;

    %[Temp] : collect Temperature data [K]
    str = 'Temperature';
    RAW{i,j}.Temp = DataLoadCol(str,dataRAW,dataNUM,4,1);
    datapoint = size(RAW{i,j}.Temp);
    Temp(:,j) = RAW{i,j}.Temp;

    %[Tamb] : collect ambient temperature [K]
    str = 'Tambient';
    RAW{i,j}.Tamb = DataLoadCol(str,dataRAW,dataNUM,1,1);
    Tamb(:,j) = RAW{i,j}.Tamb;

    %[Pdiff] : collect pressure data [Pa]
    str = 'Pressure';
    RAW{i,j}.Pdiff = DataLoadCol(str,dataRAW,dataNUM,2,1);
    Pdiff(:,j) = RAW{i,j}.Pdiff;

    %[Tdiff] : collect temperature differnet data [K]
    str = 'Delta T';
    RAW{i,j}.Tdiff = abs( DataLoadCol(str,dataRAW,dataNUM,2,0) );
    Tdiff(:,j) = RAW{i,j}.Tdiff;

    %[HTR] : collect heat transfer rate data [W]
    str = 'HTR';
    RAW{i,j}.HTR = DataLoadCol(str,dataRAW,dataNUM,1,0);
    HTR(:,j) = RAW{i,j}.HTR;

    %[LMTD] : collect log mean temperature different data [K]
    str = 'Counter';
    RAW{i,j}.LMTD = DataLoadCol(str,dataRAW,dataNUM,1,0);
    LMTD(:,j) = RAW{i,j}.LMTD;

    %[deltaT1] : collect Th,i - Tc,o data [K]
    str = 'H,in-C,out';
    RAW{i,j}.deltaT1 = DataLoadCol(str,dataRAW,dataNUM,1,0);
    deltaT1(:,j) = RAW{i,j}.deltaT1;

    %[deltaT2] : collect Th,o - Tc,i data [K]
    str = 'H,out-C,in';
    RAW{i,j}.deltaT2 = DataLoadCol(str,dataRAW,dataNUM,1,0);
    deltaT2(:,j) = RAW{i,j}.deltaT2;

    %[Aht] : collect total heat transfer area [m2]
    str = 'Total HT area [m2]';
    RAW{i,j}.Aht = DataLoadProp(str,dataRAW,dataNUM).* ones( datapoint(1),1);
    Aht(:,j) = RAW{i,j}.Aht;

    %[Acs] : collect channel crossection area [m2]
    str = 'Hot crossection area';
    RAW{i,j}.Acs = DataLoadProp(str,dataRAW,dataNUM).* ones( datapoint(1),1);
    Acs(:,j) = RAW{i,j}.Acs;

    %[Thick] : collect plate thickness[m]
    str = 'plate thickness [m]';
    RAW{i,j}.Thick = DataLoadProp(str,dataRAW,dataNUM).* ones( datapoint(1),1);
    Thick(:,j) = RAW{i,j}.Thick;

    %[UA] : collect overall heat transfer coefficeint
    str = 'UA';
    RAW{i,j}.UA = DataLoadCol(str,dataRAW,dataNUM,1,0);
    UA(:,j) = RAW{i,j}.UA;

    %[Re] : collect Overall Reynolds number
    str = 'Re';
    RAW{i,j}.Re = DataLoadCol(str,dataRAW,dataNUM,1,2);
    Re(:,j) = RAW{i,j}.Re;

    %[Re] : collect both fluid Reynolds number
    str = 'Re';
    for side = 1:2
        RAW{i,j}.Re(:,side) = DataLoadCol(str,dataRAW,dataNUM,1,side-1);
        Re(:,side,j) = RAW{i,j}.Re(:,side);
    end
    logRe = log( Re );
    RAW{i,j}.logRe = logRe;

```

```

%[Nu] : collect Nusselt number
str = 'Nu(h*d/k)';
RAW{i,j}.Nu = DataLoadCol(str,dataRAW,dataNUM,1,0);
Nu(:,j) = RAW{i,j}.Nu;
logNu = log( Nu );
RAW{i,j}.logNu = logNu;

%{
%[Pr] : collect overall Prandlr number
str = 'Pr (Cp*d.viscous/k)';
RAW{i,j}.Pr = DataLoadCol(str,dataRAW,dataNUM,1,2);
Pr(:,j) = RAW{i,j}.Pr;
%}

%[Pr] : collect both fluid Prandlr number
str = 'Pr (Cp*d.viscous/k)';
for side = 1:2
    RAW{i,j}.Pr(:,side) = DataLoadCol(str,dataRAW,dataNUM,1,side-1);
    Pr(:,side,j) = RAW{i,j}.Pr(:,side);
end
logPr = log( Pr );
RAW{i,j}.logPr = logPr;

%{
%[NuPr13] : collect Overall Nusselt/Prandlr^(1/3)
str = 'Nu/Pr^1/3';
RAW{i,j}.NuPr13 = DataLoadCol(str,dataRAW,dataNUM,1,2);
NuPr13(:,j) = RAW{i,j}.NuPr13;
%}
%[NuPr13] : collect Fluid Nusselt/Prandlr^(1/3)
str = 'Nu/Pr^1/3';
for side = 1:2
    RAW{i,j}.NuPr13(:,side) = DataLoadCol(str,dataRAW,dataNUM,1,side-1);
    NuPr13(:,side,j) = RAW{i,j}.NuPr13(:,side);
end
logNuPr13 = log( NuPr13 );
RAW{i,j}.logNuPr13 = logNuPr13;

%[Q] : collect flow rate [m3/s]
str = 'Flowrate';
RAW{i,j}.Q = DataLoadCol(str,dataRAW,dataNUM,1,1)/3600;
Q(:,j) = RAW{i,j}.Q;

%[V] : collect fluid velocity [m/s]
str = 'V';
RAW{i,j}.V = DataLoadCol(str,dataRAW,dataNUM,2,0);
V(:,j) = RAW{i,j}.V;

%[Rho] : collect density [kg/m]
str = 'Density';
RAW{i,j}.Rho = DataLoadCol(str,dataRAW,dataNUM,2,0);
Rho(:,j) = RAW{i,j}.Rho;

%[Dvis] : collect dynamics viscosity [Pa s]
str = 'Dynamic Viscosity';
RAW{i,j}.Dvis = DataLoadCol(str,dataRAW,dataNUM,2,0);
Dvis(:,j) = RAW{i,j}.Dvis;

%[DvisOA] : collect Over All dynamics viscosity [Pa s]
str = 'Dynamic Viscosity';
RAW{i,j}.DvisOA = DataLoadCol(str,dataRAW,dataNUM,1,2);
DvisOA(:,j) = RAW{i,j}.DvisOA;

%[Thdhydrau] : collect thermal hydraulic diameter [m]
str = 'Thermal Hydrualic diameter';
RAW{i,j}.ThDhydrau = DataLoadProp(str,dataRAW,dataNUM).* ones( datapoint(1),1);
ThDhydrau(:,j) = RAW{i,j}.ThDhydrau;

%[Wtdhydrau] : collect wetted hydraulic diameter [m]
str = 'Wetted Hydrualic diameter';
RAW{i,j}.WtDhydrau = DataLoadProp(str,dataRAW,dataNUM).* ones( datapoint(1),1);
WtDhydrau(:,j) = RAW{i,j}.WtDhydrau;

%[K] : collect plate Thermal conductivity [W/mK]
str = 'plate conductivity [W/mK]';
RAW{i,j}.K = DataLoadProp(str,dataRAW,dataNUM).* ones( datapoint(1),1);
K(:,j) = RAW{i,j}.K;

%[KF] : collect fluid Thermal conductivity [W/mK]
str = 'K (Therm. Conductivity)';
RAW{i,j}.KF = DataLoadCol(str,dataRAW,dataNUM,2,0);
KF(:,j) = RAW{i,j}.KF;

%[H] : collect Heat convectoin coefficient [W/m2K]
str = 'h';
RAW{i,j}.H = DataLoadCol(str,dataRAW,dataNUM,2,0);
H(:,j) = RAW{i,j}.H;

%[Rcond] : collect thermal conduction resistance [K/W]

```

```

str = 'Rcond';
RAW{i,j}.Rcond = DataLoadCol(str,dataRAW,dataNUM,1,0);
Rcond(:,j) = RAW{i,j}.Rcond;

%[Rconv] : collect thermal convection resistance [K/W]
str = 'Rconv';
RAW{i,j}.Rconv = DataLoadCol(str,dataRAW,dataNUM,2,0);
Rconv(:,j) = RAW{i,j}.Rconv;

%[R] : collect all thermal resistance [K/W]
R(:,j) = [Rconv(:,j) Rcond(:,j) Rconv(:,j)];
RAW{i,j}.R = R(:,j);

%[Cp] : collect Heat capacity at constant pressure [J/kgK]
str = 'Cp';
RAW{i,j}.Cp = DataLoadCol(str,dataRAW,dataNUM,2,0);
Cp(:,j) = RAW{i,j}.Cp;

%[Lupho] : collect upper hot side lenght [m]
str = 'Upper hot lenght [mm]';
RAW{i,j}.Lupho = DataLoadProp(str,dataRAW,dataNUM).* ones( datapoint(1),1) /1000;
Lupho(:,j) = RAW{i,j}.Lupho;

%[Lupco] : collect upper cold side lenght [m]
str = 'Upper cold lenght [mm]';
RAW{i,j}.Lupco = DataLoadProp(str,dataRAW,dataNUM).* ones( datapoint(1),1) /1000;
Lupco(:,j) = RAW{i,j}.Lupco;

%[Lloho] : collect lower hot side lenght [m]
str = 'Lower hot lenght [mm]';
RAW{i,j}.Lloho = DataLoadProp(str,dataRAW,dataNUM).* ones( datapoint(1),1) /1000;
Lloho(:,j) = RAW{i,j}.Lloho;

%[Lloco] : collect lower cold side lenght [m]
str = 'Lower cold lenght [mm]';
RAW{i,j}.Lloco = DataLoadProp(str,dataRAW,dataNUM).* ones( datapoint(1),1) /1000;
Lloco(:,j) = RAW{i,j}.Lloco;

disp(['--collect case ', caseN0str, ' complete'])

end

%% data analysis
%statistic value calculation
disp('-----')
%[case name]
CAL{i,1}.Case = sheetname{i};

%[mean]
CAL{i,1}.Temp = mean(Temp,3);
CAL{i,1}.Tdiff = mean(Tdiff,3);
%remove tamb zero
Tamb0 = Tamb(:, all( any( Tamb ), 1 ) );
CAL{i,1}.Tamb = mean(Tamb0,2);
CAL{i,1}.HTR = mean(HTR,2);
CAL{i,1}.LMTD = mean(LMTD,2);
CAL{i,1}.deltaT1 = mean(deltaT1,2);
CAL{i,1}.deltaT2 = mean(deltaT2,2);
CAL{i,1}.Aht = mean(Aht,2);
CAL{i,1}.Acs = mean(Acs,2);
CAL{i,1}.UA = mean(UA,2);
CAL{i,1}.Re = mean(Re,3);
CAL{i,1}.logRe = mean(logRe,3);
CAL{i,1}.Nu = mean(Nu,2);
CAL{i,1}.logNu = mean(logNu,2);
CAL{i,1}.Pr = mean(Pr,3);
CAL{i,1}.logPr = mean(logPr,3);
CAL{i,1}.NuPr13 = mean(NuPr13,3);
CAL{i,1}.logNuPr13 = mean(logNuPr13,3);
CAL{i,1}.Rho = mean(Rho,3);
CAL{i,1}.Dvis = mean(Dvis,3);
CAL{i,1}.DvisOA = mean(DvisOA,2);
CAL{i,1}.ThDhydrau = mean(ThDhydrau,2);
CAL{i,1}.WtDhydrau = mean(WtDhydrau,2);
CAL{i,1}.Q = mean(Q,2);
CAL{i,1}.V = mean(V,2);
CAL{i,1}.Thick = mean(Thick,2);
CAL{i,1}.K = mean(K,2);
CAL{i,1}.KF = mean(KF,3);
CAL{i,1}.H = mean(H,3);
CAL{i,1}.Rcond = mean(Rcond,2);
CAL{i,1}.Rconv = mean(Rconv,3);
CAL{i,1}.R = mean(R,3);
CAL{i,1}.Cp = mean(Cp,3);
CAL{i,1}.Lupho = mean(Lupho,2);
CAL{i,1}.Lupco = mean(Lupco,2);
CAL{i,1}.Lloho = mean(Lloho,2);

```



```
CAL{i,1}.Lloco = mean(Lloco,2);
CAL{i,1}.L = mean( [CAL{i,1}.Lupho ; CAL{i,1}.Lupco ; CAL{i,1}.Lloho ; CAL{i,1}.Lloco] );
```

```
TempM = mean(Temp,3);
TdiffM = mean(Tdiff,3);
TambM = CAL{i,1}.Tamb;
HTRM = mean(HTR,2);
LMTDM = mean(LMTD,2);
deltaT1M = mean(deltaT1,2);
deltaT2M = mean(deltaT2,2);
AhtM = mean(Aht,2);
AcSM = mean(Acs,2);
UAM = mean(UA,2);
ReM = mean(Re,3);
logReM = mean(logRe,3);
NuM = mean(Nu,2);
logNuM = mean(logNu,2);
PrM = mean(Pr,3);
logPrM = mean(logPr,3);
NuPr13M = mean(NuPr13,2);
logNuPr13M = mean(logNuPr13,2);
RhoM = mean(Rho,3);
DvisM = mean(Dvis,3);
DvisOAM = mean(DvisOA,2);
ThDhydrauM = mean(ThDhydrau,2);
WtDhydrauM = mean(WtDhydrau,2);
QM = mean(Q,2);
VM = mean(V,2);
ThickM = mean(Thick,2);
KM = mean(K,2);
KFM = mean(KF,2);
HM = mean(H,3);
RcondM = mean(Rcond,2);
RconvM = mean(Rconv,3);
RM = mean(R,3);
CpM = mean(Cp,3);
LuphoM = mean(Lupho,2);
LupcoM = mean(Lupco,2);
LlohoM = mean(Lloho,2);
LlocoM = mean(Lloco,2);
LM = CAL{i,1}.L;
```

```
%% error calculation
disp('Calculating Data Error');
sideSTR = {'hot','cold'};
%[TempError] : temperature error calculation
disp('--Calculating Temperature Error');
for port = 1:4
    X = Temp(:,port,:);
    x = reshape(X,[],subcase);
    TError = PrecBiasError(x,0.05);
    TempError(:,port) = TError;
    ERR{i,1}.Temp(:,port) = TError;
end
clearvars X x;
```

```

%[TdiffError] : temperature different error calculation
disp('--Calculating Temp different Error');
for side = 1:2
    X = Tdiff(:,side,:);
    x = reshape(X,[],subcase);
    Tdifferror = PrecBiasError(x,0.05);
    TdiffError(:,side) = Tdifferror;
    ERR{i,1}.Tdiff(:,side) = Tdifferror;
    %Tdifferror(:) = sqrt( TempError(:,(side*2)-1).^2 + TempError(:,side*2).^2 );
    %TdiffError(:,side) = Tdifferror;
end
clearvars X x;
```

```

%[QError] : calculate flow rate error
disp('--Calculating Flow rate Error');
QError = 0.01 .* ones( datapoint(1),1 );
ERR{i,1}.Q = QError;
QerrorP = 0.017; %propotion from flow rate
clearvars X x;
```

```

%[RhoError] : Rho error calculation
disp('--Calculating Density Error');
for side = 1:2
    X = Rho(:,side,:);
    x = reshape(X,[],subcase);
    Rhoerror = PrecBiasError(x,0);
    RhoError(:,side) = Rhoerror;
    ERR{i,1}.Rho(:,side) = Rhoerror;
end
```

```

clearvars X x;

%[CpError] : Heat capacity at constant pressure error calculation
disp('--Calculating Heat Capacity Error');
for side = 1:2
    X = Cp(:,side,:);
    x = reshape(X,[],subcase);
    Cperror = PrecBiasError(x,0);
    CpError(:,side) = Cperror;
    ERR{i,1}.Cp(:,side) = Cperror;
end
clearvars X x;

%[HTRError] : Heat transfer rate error calculation
disp('--Calculating Heat Transfer Rate Error');
syms HTRfunc( Qvar , Rhovar , Cpvar , DeltaTvar )
HTRfunc( Qvar , Rhovar , Cpvar , DeltaTvar ) = Qvar * Rhovar * Cpvar * DeltaTvar;
var = { 'Qvar' , 'Rhovar' , 'Cpvar' , 'DeltaTvar' };

M(:,1) = QM;
M(:,2) = RhoM(:,2);
M(:,3) = CpM(:,2);
M(:,4) = TdiffM(:,2);

E(:,1) = QError;
E(:,2) = RhoError(:,2);
E(:,3) = CpError(:,2);
E(:,4) = TdiffError(:,2);

HTRError = ErrFunction(HTRfunc,var,M,E);
ERR{i,1}.HTR = HTRError;
clearvars M E;

%[deltaT1Error] : Th,i - Tc,o error calculation
disp('--Calculating Th,i - Tc,o Error');
deltaT1Error = PrecBiasError(deltaT1,0.0707);
ERR{i,1}.deltaT1 = PrecBiasError(deltaT1,0.0707);
clearvars X x;

%[deltaT2Error] : Th,o - Tc,i error calculation
disp('--Calculating h,o - Tc,i Error');
deltaT2Error = PrecBiasError(deltaT2,sqrt(0.0707));
ERR{i,1}.deltaT2 = PrecBiasError(deltaT2,sqrt(0.0707));
clearvars X x;

%[LMTDError] : Log mean temperature different error calculation
disp('--Calculating Log Mean Temperature Different Error');
%lmtd = deltaT2 - deltaT1 / log(deltaT2/deltaT1)
%by : deltaT2 = Th,o - Tc,i, deltaT1 = Th,i - Tc,o
%check if deltaT2 - deltaT1 = 0
LMTDcheck = abs( deltaT2M - deltaT1M );
icheck = find( LMTDcheck <= 10^-13 );
%replace troubled value with +100 and will be corrected later
if length(icheck) ~= 0
    for index =1:length(icheck)
        deltaT1M( icheck(index) ) = deltaT1M( icheck(index) )+10000000;
    end
end
%LMTDError = sqrt( [ df/dt1 * et1 ]^2
%                + [ df/dt2 * et2 ]^2 )

%create function
syms LMTDfunc(t1,t2)
LMTDfunc(t1,t2) = ( t2 - t1 ) / log(t2/t1);
var = { 't1' , 't2' };
M(:,1) = deltaT1M;
M(:,2) = deltaT2M;
E(:,1) = deltaT1Error;
E(:,2) = deltaT2Error;

LMTDError = ErrFunction(LMTDfunc,var,M,E);
ERR{i,1}.LMTD = LMTDError;
clearvars M E;

%check over value
if length(icheck) ~= 0
    for index=1:length(icheck)
        LMTDError( icheck(index) ) = 0.5*( sum( TempError( icheck(index),: ),2 ) );
        ERR{i,1}.LMTD( icheck(index) ) = 0.5*( sum( TempError( icheck(index),: ),2 ) );
    end
end
LMTDErrorP = ( LMTDError./LMTDM )*100;
clearvars X x;

%[AhtError] : Total heat transfer Area error
disp('--Calculating Heat transfer Area Error');
AhtError = 0.0005 .* ones( datapoint(1),1 );
ERR{i,1}.Aht = AhtError;

```

```

%[AcError] : Channel cross section Area error
disp('--Calculating Channel Cross Section Area Error');
AcError = 0.0005 .* ones( datapoint(1),1 );
ERR{i,1}.Ac = AcError;

%[ThickError] : Thickness error
disp('--Calculating Thickness Error');
ThickError = 0.0001 .* ones( datapoint(1),1 );
ERR{i,1}.Thick = ThickError;

%[ThDhydrauError] : thermal hydraulic diameter error
disp('--Calculating Thermal Hydraulc diameter Error');
ThDhydrauError = 0.000005 .* ones( datapoint(1),1 );
ERR{i,1}.ThDhydrau = ThDhydrauError;

%[WtDhydrauError] : wetted hydraulic diameter error
disp('--Calculating Wetted Hydraulc diameter Error');
WtDhydrauError = 0.000005 .* ones( datapoint(1),1 );
ERR{i,1}.WtDhydrau = WtDhydrauError;

%[DvisError] : dynamic viscosity error calculation
disp('--Calculating Dynamic Viscosity Error');
for side = 1:2
    X = Dvis(:,side,:);
    x = reshape(X,[],subcase);
    Dviserror = PrecBiasError(x,0);
    DvisError(:,side) = Dviserror;
    ERR{i,1}.Dvis(:,side) = Dviserror;
end
clearvars X x;

%[DvisOError] : Overall dynamic viscosity error calculation
disp('--Calculating Overall Dynamic Viscosity Error');
DvisOError = PrecBiasError(DvisOA,0);
ERR{i,1}.DvisOA = DvisOError;

%[KError] : Plate thermal conductivity error calculation
disp('--Calculating Plate Thermal Conductivity Error');
KError = PrecBiasError(K,0);
ERR{i,1}.K(:,side) = KError;

%[KFError] : fluid Thermal conductivity error calculation
disp('--Calculating Fluid Thermal Conductivity Error');
for side = 1:2
    X = Dvis(:,side,:);
    x = reshape(X,[],subcase);
    KFError = PrecBiasError(x,0);
    KFError(:,side) = KFError;
    ERR{i,1}.KF(:,side) = KFError;
end
clearvars X x;

%[ReError] : Fluid Reynolds number error calculation
disp('--Calculating Fluid Reynolds Number Error');
%create function
syms Refunc(Rhovar, Qvar, WtDhydrauvar, Avar, dvisvar)
Refunc(Rhovar, Qvar, WtDhydrauvar, Avar, dvisvar) = (Rhovar*Qvar*WtDhydrauvar)/(Avar*dvisvar);
logRefunc(Rhovar, Qvar, WtDhydrauvar, Avar, dvisvar) =
log((Rhovar*Qvar*WtDhydrauvar)/(Avar*dvisvar));
var = {'Rhovar', 'Qvar', 'WtDhydrauvar', 'Avar', 'dvisvar'};
for side = 1:2
    disp(['-for ',sideSTR{side},' side']);
    M(:,1) = RhoM(:,side);
    M(:,2) = QM;
    M(:,3) = WtDhydrauM;
    M(:,4) = AcSM;
    M(:,5) = DvisM(:,side);

    E(:,1) = RhoError(:,side);
    E(:,2) = QError;
    E(:,3) = WtDhydrauError;
    E(:,4) = AcError;
    E(:,5) = DvisError(:,side);

    ReError(:,side) = ErrFunction(Refunc,var,M,E);
    disp(' >Calculating log Error');
    logReError(:,side) = ErrFunction(logRefunc,var,M,E);
end
ERR{i,1}.Re = ReError;
ERR{i,1}.logRe = logReError;
clearvars M E;

%[UError] : Overall heat transfer coefficient error calculation
disp('--Calculating Overall Heat Transfer Coefficient Error');
syms Ufunc( HTRvar, LMTDvar )
Ufunc( HTRvar, LMTDvar ) = HTRvar / LMTDvar ;
var = {'HTRvar', 'LMTDvar'};

```

```

M(:,1) = HTRM;
M(:,2) = LMTDM;

E(:,1) = HTRError;
E(:,2) = LMTDError;

UAError = ErrFunction(UAfunc,var,M,E);
ERR{i,1}.UA = UAError;
clearvars M E;

%[NuError] : Nusselt number error calculation
disp('--Calculating Nusselts Number Error');
%create function
syms Nufunc( ThDhydrauvar , Khvar , Kcvar , LMTDvar , Ahtvar , HTRvar , Thickvar , Kvar )
Nufunc( ThDhydrauvar , Khvar , Kcvar , LMTDvar , Ahtvar , HTRvar , Thickvar , Kvar ) =
ThDhydrauvar * ((1/Khvar)+(1/Kcvar)) / ( ( LMTDvar * Ahtvar / HTRvar ) - ( Thickvar - Kvar ) );
logNufunc( ThDhydrauvar , Khvar , Kcvar , LMTDvar , Ahtvar , HTRvar , Thickvar , Kvar ) =
log(ThDhydrauvar * ((1/Khvar)+(1/Kcvar)) / ( ( LMTDvar * Ahtvar / HTRvar ) - ( Thickvar - Kvar )
));
var = { 'ThDhydrauvar' , 'Khvar' , 'Kcvar' , 'LMTDvar' , 'Ahtvar' , 'HTRvar' , 'Thickvar' ,
'Kvar' };

M(:,1) = ThDhydrauM;
M(:,2) = KFM(:,1);
M(:,3) = KFM(:,2);
M(:,4) = LMTDM;
M(:,5) = AhtM;
M(:,6) = HTRM;
M(:,7) = ThickM;
M(:,8) = KM;

E(:,1) = ThDhydrauError;
E(:,2) = KFError(:,1);
E(:,3) = KFError(:,2);
E(:,4) = LMTDError;
E(:,5) = AhtError;
E(:,6) = HTRError;
E(:,7) = ThickError;
E(:,8) = KError;

NuError = ErrFunction(Nufunc,var,M,E);
disp(' >Calculating log Error');
logNuError = ErrFunction(logNufunc,var,M,E);
ERR{i,1}.Nu = NuError;
ERR{i,1}.logNu = logNuError;
clearvars M E;

%[HError] : Heat convection error calculation
disp('--Calculating Heat Convection Coefficient Error');
%create function
syms Hfunc( Nuvar , Kvar , ThDhydrauvar )
Hfunc( Nuvar , Kvar , ThDhydrauvar ) = Nuvar * Kvar / ThDhydrauvar;
var = { 'Nuvar' , 'Kvar' , 'ThDhydrauvar' };
for side = 1:2
disp([' -for ',sideSTR{side},' side']);
M(:,1) = NuM;
M(:,2) = KFM(:,side);
M(:,3) = ThDhydrauM;

E(:,1) = NuError;
E(:,2) = KFError(:,side);
E(:,3) = ThDhydrauError;

HError(:,side) = ErrFunction(Hfunc,var,M,E);
end
ERR{i,1}.H = HError;
clearvars M E;

%[RcondError] : Thermal conduction resistance error calculation
disp('--Calculating Thermal Conduction Resistance Error');
%create function
syms Rcondfunc( Thickvar , Kvar , Ahtvar )
Rcondfunc( Thickvar , Kvar , Ahtvar ) = Thickvar/Kvar/Ahtvar;
var = { 'Thickvar' , 'Kvar' , 'Ahtvar' };
M = [ThickM,KM,AhtM];
E = [ThickError,KError,AhtError];
RcondError = ErrFunction(Rcondfunc,var,M,E);
ERR{i,1}.Rcond = RcondError;
clearvars M E;

%[RconvError] : Thermal convection resistance error calculation
disp('--Calculating Thermal Convection Resistance Error');
%create function
syms Rconvfunc( Hvar , Ahtvar )
Rconvfunc( Hvar , Ahtvar ) = 1/Hvar/Ahtvar;
var = { 'Hvar' , 'Ahtvar' };

```

```

for side = 1:2
    disp(['          -for ',sideSTR{side},' side']);
    M = [HM(:,side),AhtM];
    E = [HError(:,side),AhtError];
    RconvError(:,side) = ErrFunction(Rconvfunc,var,M,E);
end
ERR{i,1}.Rconv = RconvError;
clearvars M E;

%[RError] : collect thermal resistance error
RError = [RconvError(:,1) RcondError RconvError(:,2)];
ERR{i,1}.R = RError;

%[PrError] : Fluid Prandtl number error calculation
disp('--Calculating Prandtl Number Error');
%create function
syms Nufunc( Cpvar , Dvisvar , Kvar )
Prfunc( Cpvar , Dvisvar , Kvar ) = ( Cpvar * Dvisvar )/ Kvar;
logPrfunc( Cpvar , Dvisvar , Kvar ) = log(( Cpvar * Dvisvar )/ Kvar );
var = { 'Cpvar' , 'Dvisvar' , 'Kvar' };
for side = 1:2
    disp(['          -for ',sideSTR{side},' side']);
    M(:,1) = CpM(:,side);
    M(:,2) = DvisM(:,side);
    M(:,3) = KM;

    E(:,1) = CpError(:,side);
    E(:,2) = DvisError(:,side);
    E(:,3) = KError;

    PrError(:,side) = ErrFunction(Prfunc,var,M,E);
    disp('          >Calculating log Error');
    logPrError(:,side) = ErrFunction(logPrfunc,var,M,E);
end
ERR{i,1}.Pr = PrError;
ERR{i,1}.logPr = logPrError;
clearvars M E;

%[NuPr13Error] : Fluid Nusselt number / ( Prandtl number )^(1/3) error calculation
disp('--Calculating Nu/Pr^(1/3) Error');
%create function
syms NuPr13func( Nuvar , Prvar )
NuPr13func( Nuvar , Prvar ) = Nuvar / ( Prvar ^ (1/3));
logNuPr13func( Nuvar , Prvar ) = log( Nuvar / ( Prvar ^ (1/3)) );
var = { 'Nuvar' , 'Prvar' };
for side = 1:2
    disp(['          -for ',sideSTR{side},' side']);
    M(:,1) = NuM;
    M(:,2) = PrM(:,side);

    E(:,1) = NuError;
    E(:,2) = PrError(:,side);

    NuPr13Error(:,side) = ErrFunction(NuPr13func,var,M,E);
    disp('          >Calculating log Error');
    logNuPr13Error(:,side) = ErrFunction(logNuPr13func,var,M,E);
end
ERR{i,1}.NuPr13 = NuPr13Error;
ERR{i,1}.logNuPr13 = logNuPr13Error;
clearvars M E;

disp(['calculate case : ' , sheetname{icase}, ' complete' ])
disp('=====')

clearvars -except dataRAW isheet sheetname RAW CAL ERR filename;
end

%% complete
disp('Data process complete')
clearvars isheet sheetname filename dataRAW;

end

```

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